TO

MY FATHER

CERTIFICATE

This is to certify that the work on ''Design, Development and Testing of a Solar Pump'', has been carried out under my supervision and has not been submitted elsewhere for a degree.

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NOMENCLATURE

A_c	Cylinder Cross-sectional Area, cm^2
A_E	End Opening Area Of The Concentrator, cm^2
A_p	Cross-sectional Area Of The Lower Pipe, cm^2
A_T	Top Opening Surface Area Of The Concentrator, cm^2
b	Angle Of Slope, degree
CR	Concentration Ratio
d	Declination Of The Sun, degree
d_1	Entrance Aperture Of The Concentrator, cm
h	Hour Angle, degree
i	Angle Of Incidence, degree
I_{DH}	Instantaneous Value Of The Direct Radiation Rate On A Horizontal Surface, cal/cm^2-hr.
I_{DT}	Instantaneous Value Of The Direct Solar Radiation On A Tilted Surface, cal/cm^2-hr.
I_{GH}	Instantaneous Value Of The Global Radiation Rate On A Horizontal Surface, cal/cm^2-hr.
L_s	Stroke Length, cm
n	Wall Solar Azimuth, degree
N	Number Of Cycles Per Hour
Q	Discharge Of The Solar Pump, lit/hr
Q_{cl}	Heat Load On The Condenser, Kcal/hr
Q_{IT}	Total Energy Received By The Concentrator, cal/hr

S	Circumference Of The Receiver Pipe, cm
t_c	Condensate Temperature, ^oC
t_w	Temperature Of Cooling Water, ^oC
U_o	Overall Heat Transfer Coefficient, kcal/hr-m^2 ^oC
V_{cl}	Clearance Volume, cm^3
V_s	Swept Volume, cm^3
V_T	Total Volume Per Cycle, cm^3
W_F	Flow Rate Of Working Fluid, kg/hr
W_T	Pump Work, kg-m/hr
Z	Azimuth Angle, degree
β	Altitude Angle Of The Sun, degree
θ_{max}	Half Acceptance Angle, degree

ABSTRACT

A laboratory model of a new design of solar pump has been developed and tested after reviewing the existing designs. The pump works on the thermodynamic Rankine cycle using an organic fluid (Methyl Alcohol) having low boiling point. The superheated vapours of the fluid are generated in a stationary Winston concentrator specifically designed for the purpose. The vapours push a liquid piston of turpentine oil in the pump cylinder, which directly forces water from an L - shaped pipe, connected to the working cylinder, to rise in a delivery pipe through a non-return valve. The vapours, when exhausted in a condenser, produce a partial vacuum in the cylinder causing suction of fresh water from a water source through another non-return valve.

The simplified design of the pump eliminates the use of a conventional engine and the mechanical pumping device.

Experiments performed on the pumping system prove the feasibility of the idea very well. The preliminary tests in the laboratory show a discharge of about 75 litres of water per hour through a head of about 6 metres. The details of the experimental set up and the test results are given in the ensuing chapters.

CHAPTER-I

INTRODUCTION

It is well recognized that the world fossil fuel reserves are getting exhausted at a fast rate and severe shortage of these fuels is foreseen in not too distant a future; rather, it is already being experienced in some of the most developed and industrialized nations and there is a fear that it may soon create intolerable, economic and environmental problems. Furthermore, pollution caused by the use of fossil fuels has created health problems resulting in the demands for pollution-free sources of energy. Thus, there is a need to look for alternative energy sources which are virtually inexhaustible and pollution-free.

Although geothermal energy, solar energy, wind energy and energy from the tidal waves are such major alternative energy sources, we shall concentrate here on the use of SOLAR ENERGY which is received by the land areas of the globe to the extent of about 2×10^{17} KWH/yr. and is well in excess of the total energy of 5×10^{13} KWH consumed by mankind in 1970. In India, which lies in the latitude range of $7^{o}N - 37^{o}N$, the solar energy is available in abundance throughout the year (about 4·5 kwh/m^2/day in winter and about 7·0 kwh/m^2/day in summer) and it, therefore, seems to be an ideal source of energy for many of the applications.

There are three problems in harnessing the solar energy:

1. It is not available continuously throughout the day and the year in almost all parts of the world and does not exist at all during the night. This necessitates the use of some storage device.

2. The intensity of solar energy falling on earth is quite low (solar constant being 2·0 cal./cm^2-min.).

3. As of now, the initial cost of any solar device is comparatively higher as compared to the conventional devices.

Thus, in order that the solar energy be of use to mankind effectively, it must be harnessed efficiently and economically.

Knowing that the solar energy available at a particular location during the day is periodic in nature, its use for periodic needs will be greatly successful as it does not pose the complex energy storing problem. Pumping water using solar energy is one of such examples.

Although, at present, the solar pumping systems can not compete with the conventional pumps driven by the gasoline engines or electric motors, they can find a place, however, where cheap fuel is not available and the electric power can not reach at all. At such places, low capital investment, simplicity of operation and maintenance and the availability of cheap labour are required to instal solar pumping systems.

1·1 REVIEW OF PREVIOUS WORK

The early recorded evidence of a successful attempt of utilizing solar energy for water pumping, dates back to 17th century with the development of a first solar pump in France by Solomon de Caux [1] in 1615, using expansion of air, heated by solar energy in raising water. However, a systematic record of developments is available from the later half of 19th century. August Mouchat's [1] work in France between 1866-72 describes a solar pump which operates a reciprocating or rotative pump with steam engine driven by steam, generated from a conical concentrator. John Ericsson [1] in New York started working in this area in the year 1868 using series of parabolic troughs. Several successful hot air engines were developed and in 1883, an engine using steam or air as working medium to operate a 5'' diameter force pump was introduced.

First steam engine in India operated by solar heat, used for pumping purposes, is reported to have been developed in 1876 by W. Adams [1] in Bombay which used a 40 feet diameter hemispherical mirror as concentrator to generate the steam. A series of similar developments followed after it.

Development of 11 H.P. compound condensing steam engine by the party of Boston Inventor [2] utilizing a big truncated cone with a clock-operated sun tracking mechanism was reported in 1901 which, by connecting a centrifugal pump, pumped 1400 gallons/min. of water and developed 4.25 H.P.

An excellent review of the salient contribution in developing solar engine to be used for water pumping has been given by Richard C. Jordan & Warren E. Ibele [1] till 1955. It also mentions about the solar power plant developed by Shuman and Boys' and the hot air engine operating a water pump, by M.L. Ghai and M.L. Khanna at National Physical Laboratory New Delhi [1,3]. Paraboloidal concentrators of different sizes have been used to concentrate the sun energy.

The general trend in all the above mentioned works has been primarily to develop a solar engine either working on hot air cycle or using steam as a working fluid. The coupling of any conventional water pump with this engine made utilization of solar energy for water pumping purposes.

H.E. Willsie and John Boyle Jr.[1] run a 20 H.P. slide valve engine by using a low boiling point volatile working fluid like SO_2, NH_3 or ether. A centrifugal water pump was operated by this engine using SO_2 vapours which were generated in a fire tube boiler kept in shallow water basin with double glazing cover used as solar energy collector.

J.P. Girardier and H. Masson's [4] work differs in a way that it introduces a new type of pumping system. The pump can be lowered inside a well and is operated by hydraulic pulses transmitted to it through a control pipe from the hydraulic ram, run by turbine. Here also, the heat collected by flat plate collectors from the solar energy is used to generate high pressure vapours of methyl chloride which run the turbine.

F.A. Bonaventura and V.E. Plymtom [5] invented a solar pumping device working on the principle that if the temperature

of a liquid in a closed container is raised, the same liquid can be lifted to a height due to the pressure generated by its vapours. The system consisted of a concentrator with sun tracking mechanism, a lens system and four boilers. One of the boilers is preheated by the lens system. It is reheated by a concentrator which causes to generate pressure inside the boiler and, thus, the water inside the boiler is lifted. On the condensation of steam inside the boiler, the vacuum is created and this sucks water inside the boiler.

In another solar-water pumping system, developed in 1885 [1], NH_3 is boiled in a flat plate collector and pressure thus created is applied on one side of the diaphragm contained in the sphere located in the tank. The other side of the diaphragm acts as a water pump. The details of the work are not available, however, in principle, the device seems to be of significant importance due to its simplicity. It eliminates completely the need of any mechanical engine or conventional pump. On the same principle, D.P. Rao and K.S. Rao [6] have developed and tested two versions of a solar pump - the air cooled and water cooled, by selecting pentane as a working fluid. Flat plate collectors are used to generate vapours. These vapours directly press the water in a closed tank located inside the well below the level of the water table to raise it through a delivery pipe. On condensation of vapours, in the other half of the cycle, the water enters the water-tank due to its own weight.

C. West [7] presented a theoretical analysis of the fluidyne heat engine with experimental verification of the feasibility of the system using small valve-less models. The

engine works on Stirling cycle and has no moving parts. The volume change and phasing are achieved hydraulically. The hot cavity volume change leads the cold cavity by an angle less than 180^{0}. The pressure change has a component in phase with the velocity of output column which can be used to over-come viscous friction in the output tube and to pump the water. Some of the available power output is used to maintain the amplitude and phasing of the liquid against viscous losses. The maximum efficiency achieved experimentally is 0·35% with the valveless model.

1·2 PRESENT WORK

The solar pumping systems discussed so far possess either few or all of the following drawbacks:

1. use of concentrators with expensive and cumbersome sun tracking mechanism,

2. requirement of both a primemover and a pump,

3. difficult and expensive maintenance and operation of the system,

4. low Carnot cycle efficiency of the system in cases where flat plate collectors have been used,

5. bulky and complicated system and

6. high initial cost.

The present work is motivated to over-come most of the above shortcomings. This has been achieved in the following manner:

1. A stationary type of concentrator is used to collect the sun energy which requires only periodic adjustment. It is not necessary, therefore, to make it movable to track the sun.

2. The high pressure vapours of the working fluid are directly used to push the water to be lifted. The use of engine is, thus, eliminated.

3. The pump is made of G.I. pipes, fittings and valves only. This makes the system very simple in construction.

4. No moving parts are used except a small feed pump to force the working fluid in the receiver pipe. Hence, the maintenance of the system is quite inexpensive.

5. The Carnot cycle efficiency of a system increases with temperature while the collection efficiency of the concentrator decreases because the heat losses are increased. Therefore, in the proposed work, the system is chosen to operate neither at low temperatures ($<80^{\circ}C$) nor at high temperatures ($>120^{\circ}C$), but at moderate temperatures of about $100^{\circ}C$ in order to achieve the optimum efficiency of the total system.

6. The system is simple in construction and easy in operation and maintenance. It does not involve, therefore, high initial and running cost.

The schematic diagram of the proposed system is shown in figure 1·1. The vapours of the working fluid are generated in the stationary concentrator which are used to push the water to be pumped. The vapours are, then, sent to the condenser where they are condensed to create vacuum to suck fresh water.

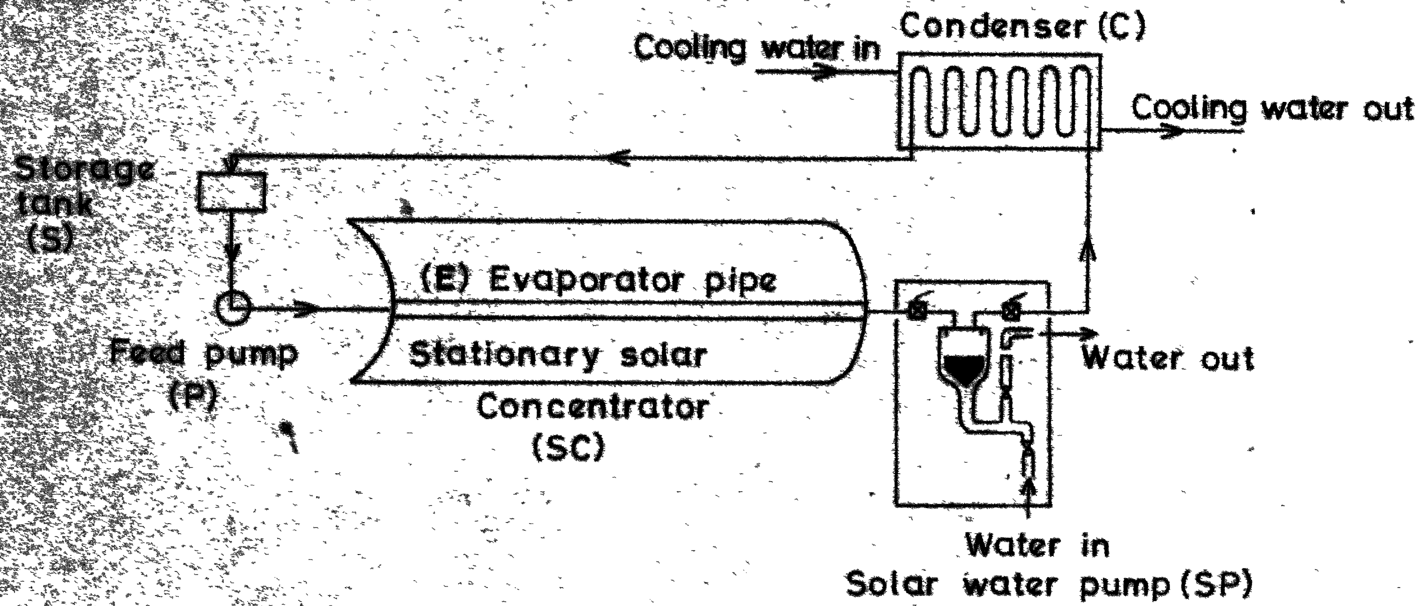

FIG. 1.1 SCHEMATIC DIAGRAM OF THE SOLAR PUMPING SYSTEM

The condensate of the working fluid is collected in a storage tank from where it is pumped back to the concentrator by means of a feed pump. The details of the fabrication and working of various components of the pumping system are explained in the text.

CHAPTER - II

DESIGN ANALYSIS AND THE EXPERIMENTAL SET-UP

A laboratory model of a solar pumping system has been developed which consists of the following main components:

(a) Solar Pump
(b) Condenser
(c) Storage tank
(d) Feed Pump and
(e) Stationary concentrator

A detailed description of the design analysis and fabrication of the various components is given below.

2·1 SOLAR PUMP

As shown in figure 2·1, the pump consists of a vertical cylinder (1) of G.I. pipe, the lower end of which is connected to a smaller diameter G.I. pipe (3) by means of a reducer (2). The other end of the pipe (3) is joined to a tee (4) with the help of an elbow and a nipple. The other two openings of the tee are connected to the delivery and suction pipes of the pump with the help of nipples, elbows and reducers through two non-return valves (5) and (6), on the delivery and the suction sides, respectively.

A float valve (7), made of circular hollow wooden box enclosing thermocoal inside its cavity and finished plane surface on its top, is inserted inside the pump cylinder. The lower cone of the valve is covered with an aluminium cup which is machined on the outer tapered surface. It rests on the tapered aluminium seat (9) fitted inside the

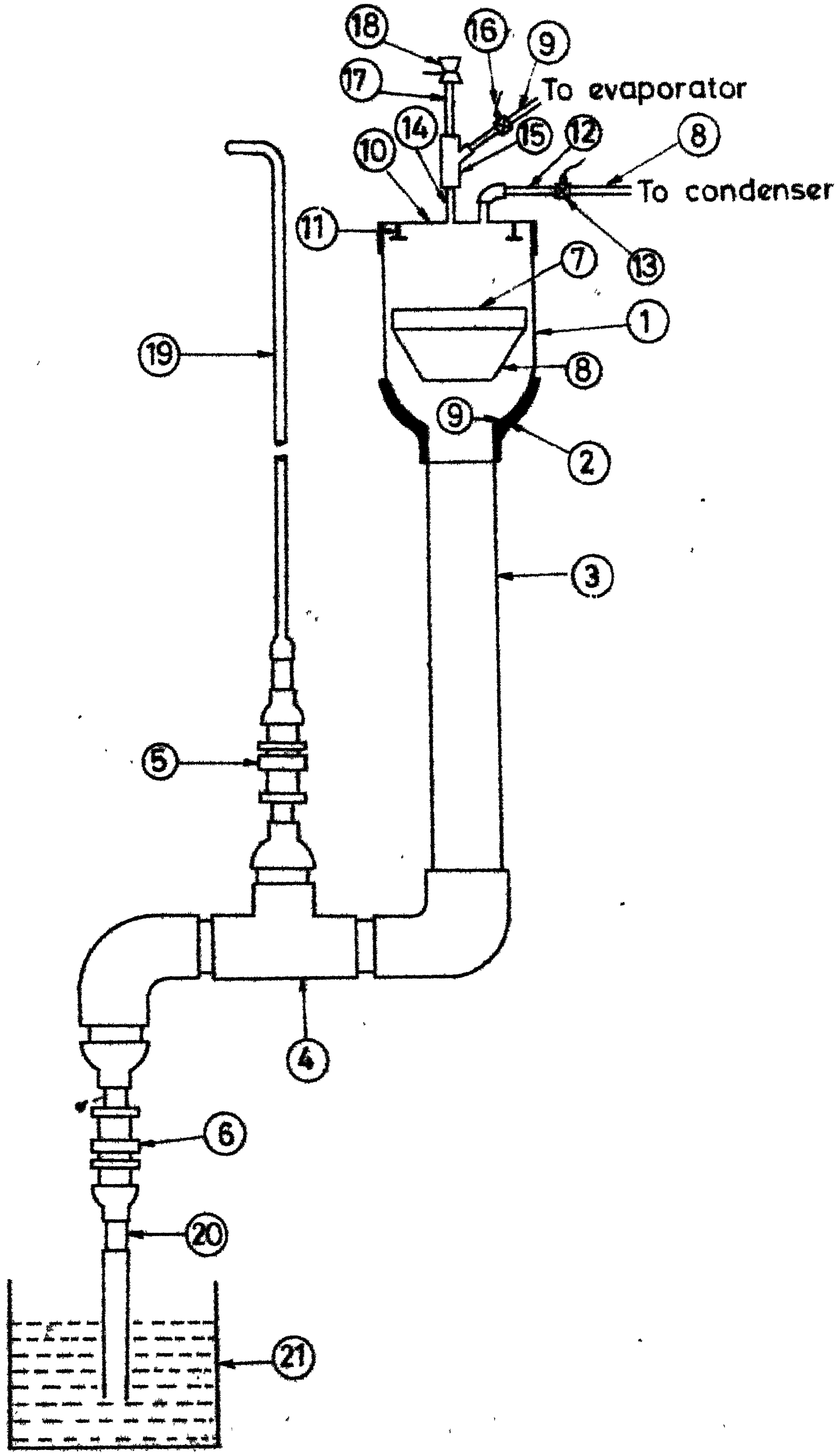

FIG. 2.1 SOLAR PUMP

reducer (2), and provides a leak proof joint for any gas or vapour going to the pipe (3) from the cylinder (1). The float valve (7) floats on a liquid piston of turpentine oil inside the pump cylinder. The turpentine oil is used as a liquid piston because it prevents mixing of working fluid vapours with the water. Also, the turpentine oil itself is immiscible with water and the working fluid.

The upper end of the cylinder is covered by a threaded cover cap (10) having a circular G.I. ring (11) welded inside it. The projected end of the ring is machine-finished which acts as a seat for the upper plane finished surface of the float valve. This provides a leak-proof contact preventing any liquid to go upward into the space, enclosed by the welded G.I. ring in the cover cap and the top surface of the float valve, from where it could have been, otherwise, sucked up into the condenser through the exhaust pipe (12). The exhaust pipe (12) is connected to the condenser through a hand operated valve (13) which is opened during the suction stroke. A G.I. pipe (14) is welded at the centre of the top cover plate of the cylinder to lead to the concentrator receiver through a tee (15) and a hand operated valve (16). A side tube (17) is provided at the tee (15) and an ordinary valve (18) is fixed at the end of the tube. This helps in releasing any gases from the cylinder while priming the solar pump.

The non-return valve (5) in the delivery pipe allows only the upward flow of water through the delivery pipe (19) and checks the downward flow. The similar non-return valve (6) on the suction side allows the entry of water from the tank (21) through the suction pipe (20) and checks

its back flow.

As explained above, there are 20 various parts of the solar pump to be designed and fabricated. While, most of the parts fall under the category of pipe and fittings and do not need precise design criteria, the pump cylinder and the pipe attached to its lower end play an important role in the pump functioning. It is, therefore, necessary to give a systematic design analysis of these two parts.

Since, only a laboratory model of the pump was proposed to be developed, the discharge capacity of the pump was chosen to be low (between 500-600 lit/hr, say 550 lit/hr) against a head of 15 meters (10m discharge head and 5m suction head), working at 15 cycles/min.*

For a single acting positive displacement type pump, the discharge is fixed by its swept volume and the number of cycles per unit time. Thus,

$$Q = \frac{A_c \times L_s \times N}{1000} \qquad (2.1)$$

Where,

Q = discharge of the pump in lit/hr

A_c = cylinder cross-sectional area in cm^2

L_s = stroke length in cm

N = number of cycles per hour

From the above relation, the swept volume is given by

$$A_c \times L_s = \frac{Q \times 1000}{N} \ cm^3$$

$$= \frac{550 \times 1000}{15 \times 60} = 611.11 \ cm^3$$

A 4'' diameter G.I. pipe was chosen for the pump cylinder which is easily available in the local market. Its actual diameter was 10·2 cm.

The area of cross-section of the pump cylinder

$$= \frac{\pi}{4} (10 \cdot 2)^2 \text{ cm}^2$$

$$= 81 \cdot 74 \text{ cm}^2$$

Hence, the length of the stroke

$$= \frac{611 \cdot 11}{81 \cdot 74} \text{ cm} = 7 \cdot 5 \text{ cm.}$$

The stroke length being small, the heat losses by convection due to the reciprocating motion of the turpentine oil column in the cylinder are reduced. Also the smaller the stroke length, the lesser is the possibility of experiencing turbulent motion of the turpentine oil. This avoids mixing of turpentine oil with water.

The length of the pump cylinder has been taken as 10 cm, somewhat more than the stroke length, to accomodate the float valve, the upper ring and a seat at the lower end.

At the lower end of the pump cylinder, a $2\frac{1}{2}$'' diameter (actual internal diameter = 6·90 cm) standard G.I. Pipe was connected by means of a reducer (4'' Ø x $2\frac{1}{2}$'' Ø). The use of the reducer facilitates fixing the seat for the float

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*· It was expected that 1 cycle may be completed in about 4 seconds by using hand operated valves.

valve and, at the same time, streamlining the flow of fluid. In order to determine the length of the lower pipe connected to the pump cylinder, the area of cross-section A_p of this pipe is given by

$$A_p = \frac{\pi}{4} (6{\cdot}90)^2 \quad cm^2$$
$$= 37{\cdot}40 \ cm^2$$

The swept volume being 611·11 cm^3, the length upto which the turpentine oil may be pushed by the working fluid vapours, from the cylinder to the lower pipe, is given by

$$L_T = \frac{611{\cdot}11}{37{\cdot}4} \quad cm \quad = \quad 16{\cdot}34 \ cm$$

Since, a liquid piston of turpentine oil is used which occupies a length of 16·34 cm of the lower pipe at the end of the downward stroke, there is a possibility of the turpentine oil getting discharged with the water if the lower pipe length is also taken as 16·34 cm. In order to avoid this difficulty, the length of the lower pipe is taken as 35 cm i.e. about two times the length L_T.

The dimensions of the remaining parts have been chosen on the basis of convenience and availability.

The following table No.2·1 gives the dimensions of the various parts.

Part No.	Name of the Part	Dimensions
1	Pump cylinder	4'' Ø, length 10 cm
2	Reducer	4'' Ø x $2\frac{1}{2}$'' Ø
3	Lower pipe	$2\frac{1}{2}$'' Ø, length 35 cm long
4	Tee	$2\frac{1}{2}$'' Ø
5	Non-return valve	1'' Ø
6	Non-return valve	1'' Ø
7	Float valve	4'' Ø wooden valve
8	Aluminium cup	$3\frac{1}{2}$'' x 2'', height 2''
9	Aluminium seat	3'' Ø with 30° tapered
10	Cover cap	4'' Ø socket
11	Top ring	$2\frac{1}{2}$'' Ø
12	Vapour outlet	1/2'' Ø
13	Ball valve	1/2'' Ø
14	Vapour inlet	1/2'' Ø
15	Tee	1/2'' Ø
16	Ball valve	1/2'' Ø
17	Side tube	1/2'' Ø
18	Ordinary valve	1/2'' Ø
19	Delivery pipe	1/2'' Ø, length 18.5 ft.
20	Suction pipe	1/2'' Ø, length 2.0 ft.

2.2 CONDENSER

The condenser used in the system consists of a tank made of a 24 gauge G.I. sheet and open at the top. It has an inlet and an outlet for circulating the fresh water. The inlet is just above the bottom of the tank and the outlet at a level higher than the inlet, so that the cold water entering the inlet comes in contact with the copper coil and gets heated. The hot water being lighter than the cold water, rises up and goes out of the tank through the outlet. The copper tube, bent to the shape as shown in figure 2.2, is inserted inside the G.I. tank. The plane of the coil is inclined, with the horizontal, at a small angle, so that the inlet of the vapours from the solar pump cylinder to the coil is at a higher level than the outlet of the condensate. This facilitates the flow of the condensate to the storage tank. The G.I. tank is kept on a frame made of slotted angle iron.

Although, in the experiment, the cooling water is circulated by a TULLU pump, the water lifted by the solar pump may itself be used to pass through the condenser to work as cooling water. Thus, the use of the circulating pump may be avoided.

The most important aspect in the design of a condenser is to estimate the length of the condensing coil used. This can be done by first knowing how much heat load the condenser has to encounter.

To determine the heat load on the condenser, we first calculate the flow rate of the working fluid (methyl alcohol)*. in the system.

*. Thermodynamic and physical properties of various organic and

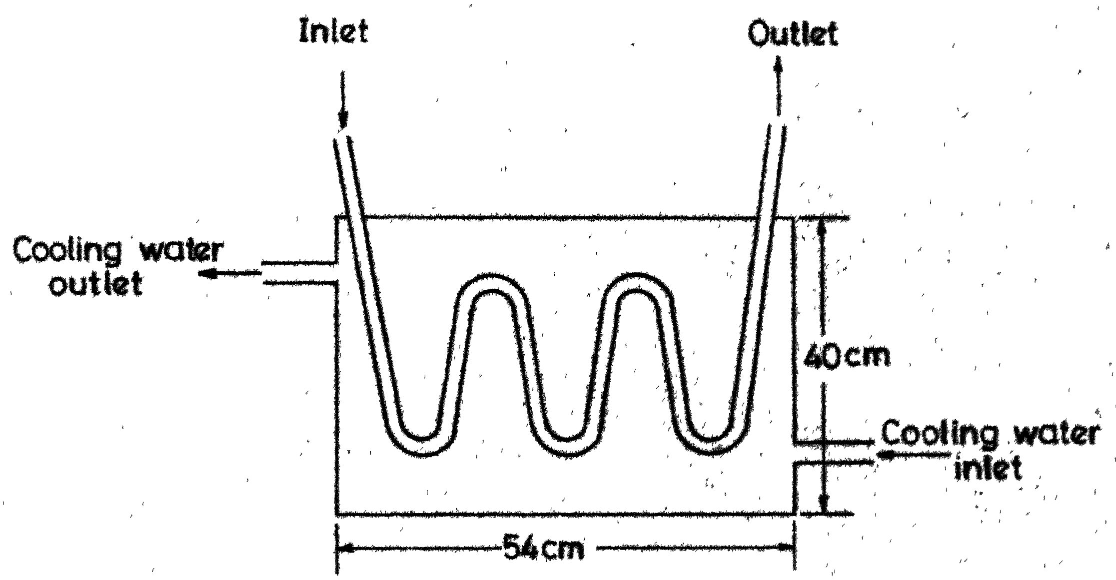


FIG. 2.2 CONDENSER

~~The total volume occupied by methyl alcohol vapours,~~

inorganic fluids were studied as given in the table below

S.No.	Name	Boiling point °C	Melting point °C	Solubility in Water	Solubility in Turpentine oil
1.	Methanol (methyl alcohol)	64.65	-97.8	soluble	insol.
2.	Isobutyl nitrite	67.00	—	soluble	insol.
3.	Isopropyl ether	67.5	-60	insol.	soluble
4.	n-Hexane	69.00	-94.3	insol.	soluble
5.	Ethanol (ethyl alcohol)	78.50	-117.3	soluble	insol.

In the present experiment, we have chosen methyl alcohol as the working fluid because of its having a moderate boiling temperature (at atmospheric pressure) and other properties suited to our design requirements. Also, it is cheap and easily available in the local market.

The total volume occupied by methyl alcohol vapours, in one cycle, at the end of the expansion stroke is given by,

$$V_T = V_s + V_{cl} \qquad (2\cdot2)$$

where

V_T = total volume per cycle

V_s = swept volume

V_{cl} = clearance volume

The swept volume was fixed to be 611·11 cc as explained in article 2·1. The clearance volume comprises three volumes i.e.

$$V_{cl} = V_i + V_o + V_e$$

where

V_i = Volume of the inlet pipe between the valves (16), (18) and the pump cylinder entry

= cross-sectional area x length of the total pipe

$= \frac{\pi}{4} (1\cdot27)^2 \times 20 \text{ cm}^3$

$= 25\cdot4 \text{ cm}^3$

V_o = volume of the outlet pipe between the valve (13) and the exit of the pump cylinder

= cross-sectional area x length of the pipe

$= \frac{\pi}{4} (1\cdot27)^2 \times 15\cdot5 \text{ cm}^3$

$= 19\cdot65 \text{ cm}^3$

V_e = volume enclosed between the cover cap and the top of the float valve.

= [cross-sectional area of pump cylinder-area of m.s. ring] x height of the m.s. ring(11)

$$= \frac{\pi}{4}(1020)^2 - \frac{\pi}{4}(7{\cdot}6)^2 - \frac{\pi}{4}(6{\cdot}9)^2 \times 0{\cdot}6 \quad cm^3$$

$$= 45{\cdot}4 \ cm^3$$

$$\therefore V_{c1} = (25{\cdot}4 + 19{\cdot}65 + 45{\cdot}4) \ cm^3 = 90{\cdot}45 \ cm^3$$

$$V_T = (611{\cdot}11 + 90{\cdot}45) \ cm^3$$
$$= 701{\cdot}56 \ cm^3$$

It is assumed that the conditions of the vapours at the cylinder outlet, and, hence, at the condenser inlet, are the same as those at the inlet of the cylinder. The vapours of the working fluid (methyl alcohol) are required to be at a temperature of 100°C (90°C saturation temperature + 10°C super heat) in order to achieve the vapour pressure equal to 25 meters of water column (absolute).

The specific volume of the saturated methyl alcohol vapours at 90°C is 0·345 m^3/kg (from the standard tables). In the superheated state of the methyl alcohol vapours, the behaviour of the vapours may be assumed to be that of the ideal gas.

With this assumption, the specific volume of the methyl alcohol vapours, in the superheated state, may be calculated by

$$\frac{P_1 v_1}{T_1} = \frac{P_2 v_2}{T_2}$$

where,

P_1 and P_2 are the pressures at the saturated and superheated states, respectively.

T_1 = absolute temperature of the saturated vapours = $(273 + 90)^{\circ}K = 363^{\circ}k$

T_2 = absolute temperature of the superheated vapours

$= (273 + 100)^{o}k = 373^{o}k$

V_1 = specific volume of the saturated vapours

$= 0.345\ m^3/kg$

V_2 = specific volume of the superheated vapours

Since, the vapours are retained in the same receiver pipe, both at the saturated and the superheated states, the pressure

$$P_1 = P_2$$

$$\therefore\ V_2 = \frac{T_2}{T_1} V_1 = \frac{373 \times 0.345}{363}\ m^3/kg$$

$$= 0.354\ m^3/kg$$

Hence, the flowrate of the methyl alcohol in the system is given by

$$W_F = \frac{\text{Total volume per cycle x Number of cycles per unit time}}{\text{specific volume of the working fluid.}}$$

$$= \frac{V_T \times N}{V_2}$$

$$= \frac{701.56 \times 10^{-6} \times 15 \times 60}{0.35}\ kg/hr$$

$$= 1.78\ kg/hr.$$

Since, the suction head has been chosen as 5 meter of water column, it is necessary that the vacuum created

in the condenser be more than 5 m. of water column. It can be easily seen that this requirement is fulfilled if we keep the condensate temperature as 40°C. The saturation pressure of the methyl alcohol at 40°C is 260 mm of Hg.[8] Hence, the vacuum created inside the condenser is (760-260 = 500) mm of Hg. or 6.58 meter of water column.

The cooling water for the condenser is available at a temperature of about 35°C or below. To be on the safer side, we assume it to be 35°C. Thus, by keeping the condensate temperature 40°C., a temperature difference of 5°C (40°C - 35°C) is available for the heat transfer to take place in the condenser.

The heat to be removed in the condenser
= sensible heat of the superheated methyl alcohol vapours + latent heat of the condensation + sensible heat of the liquid methyl alcohol

i.e. $$Q_{cl} = W_F \times C_{ps} \times (t-t_s) + W_F L + W_F C_{pw}(t_s-t_c) \quad (2.3)$$

Where,

Q_{cl} = heat load on the condenser,

C_{ps} = mean specific heat of the superheated vapours of methyl alcohol,

t = temperature of the superheated vapours of methyl alcohol,

t_c = condensate temperature,

C_{pw} = mean specific heat of the liquid methyl alcohol

$$\therefore \quad Q_{cl} = 1.78 \times 0.80 \times (100-90) + 1.78 \times 256 + 1.78 \times 0.645 \times (90-40)$$

$$= 526.75 \text{ Kcal/hr.}$$

The heat load on the condenser may also be expressed as

$$Q_{cl} = U_o A_o (t_c - t_w) \qquad (2.4)$$

where,

U_o = overall heat transfer coefficient based on the outer diameter of the copper tube,

A_o = outer surface area of the copper tube,

t_w = temperature of cooling water.

The value of the overall heat transfer coefficient U_o has been taken [9] as 250 BTU/hr.ft^2°F (=1220 K.cal/hr.ft^2°F) for methyl alcohol as hot fluid and water as cold fluid. Also it is a practice to use 5/8'' diameter copper tube in small condensers.

$$\therefore \quad A_o = \frac{Q_{cl}}{U_o(t_c - t_w)}$$

$$= \frac{526.75}{1220 \times (40-35)} \text{ m}^2$$

$$= 8.65 \times 10^{-2} \text{ m}^2$$

$$= \pi \, d_{co} \, l$$

where,

d_{co} = outer diameter of the copper tube

l = length of the condenser coil

$$\therefore \quad l = \frac{A_o}{\pi d_{co}}$$

$$= \frac{8 \cdot 65 \times 10^{-2}}{\pi \times 1 \cdot 6 \times 10^{-2}} \quad m$$

$$= 172 \quad cm.$$

The length of the copper tube was taken as 175 cm and bent to the shape as shown in figure 2.2. The dimensions of the G.I. tank, in which the coil is inserted to form the condenser were taken as 54 cm x 40 cm x 24 cm such that the coil may be fixed in it without any inconvenience.

2.3 STORAGE TANK

This is a closed tank, made of 1/8'' thick **M.S.** plate by arc welding. The condensed methyl alcohol in the condenser is stored in the tank through an inlet valve provided at the upper portion of one of its sides. The methyl alcohol stored in the tank, goes to the feed pump through a valve controled outlet. The positions of the inlet and outlet of the storage tank for methyl alcohol have been fixed so that the following purpose is served:

(i) All the methyl alcohol inside the storage tank may be sent to the generator pipe by feed pump.

(ii) Only the liquid alcohol goes to the feed pump and not the vapours, if any, present inside the storage tank.

The storage tank is kept inclined by an angle of 15^o with the horizontal. There is one discharge valve in the bottom of the storage tank and in the lower end from which all the methyl alcohol can be taken out, if desired. Two valves provided on the top face of the storage tank facilitate pouring of methyl alcohol in the storage tank. The methyl alcohol enters through one valve and the gases inside the storage tank escape out through another valve. The size of the tank has been taken as 30 cm x 20 cm x 18 cm, so as to accomodate about 10 litres of the working fluid. The position of valves and inlet, outlet in the tank are shown in figure 2.3.

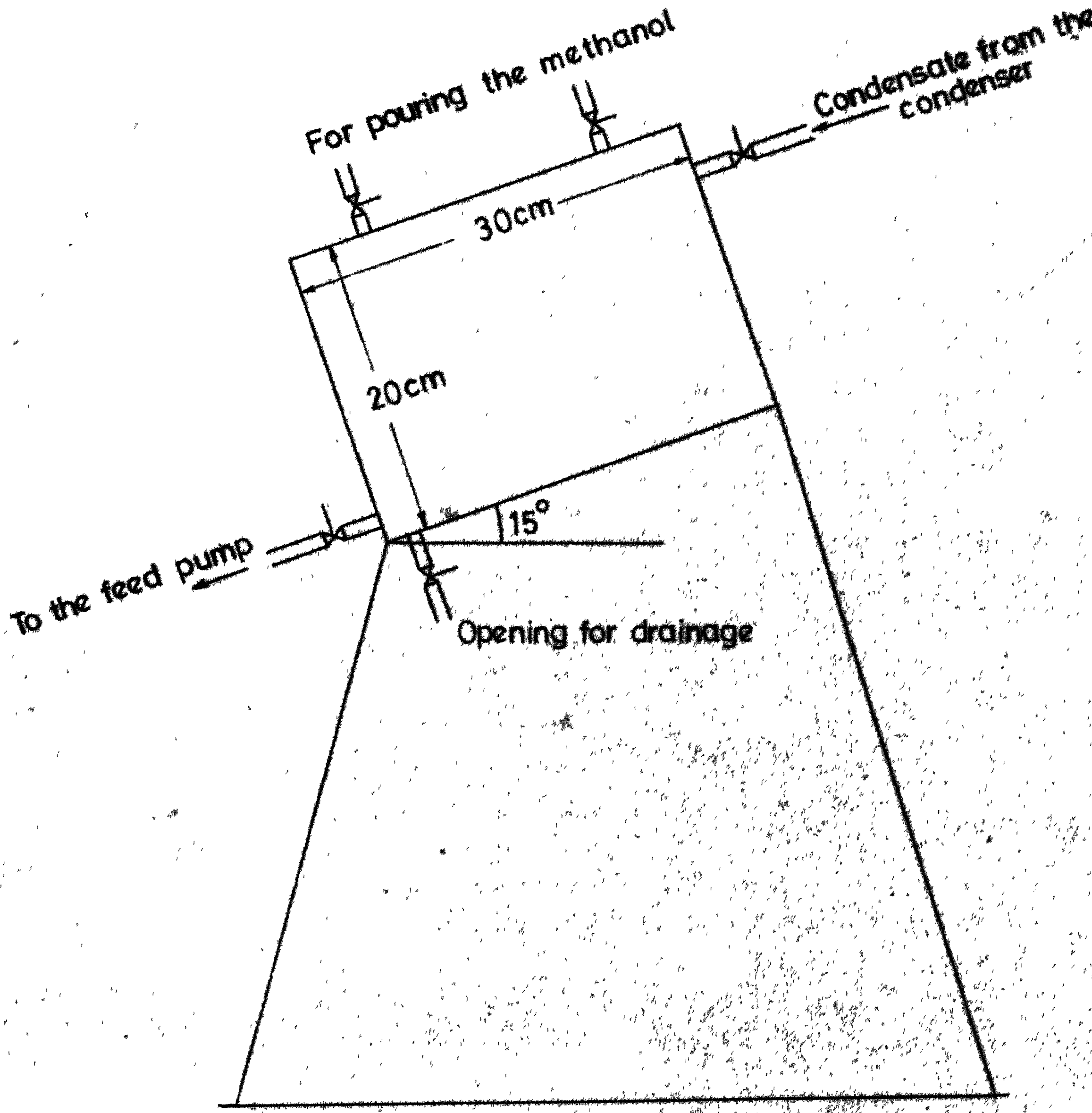


FIG. 2.3 STORAGE TANK

2·4 FEED PUMP

This is the pump used to feed the methyl alcohol to the evaporator from the storage tank. The methyl alcohol vapours coming from the solar pump are condensed in the condenser and the liquid methyl alcohol, thus obtained, is stored in the storage tank. As the system is working on the closed Rankine cycle, a feed pump is required to feed the liquid methyl alcohol to the evaporator (high pressure side) from the storage tank (low pressure side). For this purpose, a positive displacement pump with high head and low discharge is suitable, but, in our case, a TULLU pump having the following specifications is used.

Power	:	250 watt.
R.P.M.	:	5500 rpm
Discharge	:	800 lit hr.
Head	:	18·2 m of water column.

If the pressure inside the evaporator goes high, the flow of methyl alcohol (liquid and vapour both) takes place from the evaporator back to the storage tank through the TULLU pump, in situations when the pump is not working and the inlet valve to the pump cylinder is closed. Thus, the excessive pressure rise is prevented in the evaporator which, otherwise, would cause bursting of the receiver and connecting pipes on the high pressure side. This danger would not have been overcome with the positive displacement pump which does not allow back flow through it.

The mass of the methyl alcohol to be pumped from the storage tank to the evaporator is 1·78 kg/hr (see 2·2). The

methyl alcohol is in the saturated liquid state at 40°C and its density is 774 kg/m^3. Hence, the volume flow rate of the methyl alcohol to be pumped

$$= \frac{1 \cdot 78}{774} \ m^3/hr$$

$$= 2 \cdot 3 \times 10^{-3} \ m^3/hr$$

$\therefore$ the work required to pump the liquid methyl alcohol

$$W_p = \text{volume flow rate} \times (P_1 - P_2) \qquad (2 \cdot 5)$$

where,

P_1 = pressure of methyl alcohol, inside the receiver pipe, at a temperature of 90°C

P_2 = pressure of liquid methyl alcohol in the storage tank at 40°C

Using standard table,

$$W_p = 2 \cdot 3 \times 10^{-3} \ (2 \cdot 49 - 0 \cdot 342) \times 10^4 \ \frac{kg - m}{hr}$$

$$= 49 \cdot 3 \ \frac{kg - m}{hr}$$

2.5 STATIONARY CONCENTRATOR

In the present system, the modified Winston concentrator has been adopted to collect solar radiation as a heat source to vaporize the working fluid. The concentrator is a cylindrical light collector having trough-like reflecting walls. The trough walls have a specific shape which look ~~like~~ parabolic and concentrate direct radiant energy on to a tube receiver of general shape, circular, oval, rectangular and even fin-like. The tubes cross-section may have any shape provided that it is convex and symmetric about the optic axis of the concentrator. In fact, the tubes surface need not be even closed. The requirement that the cross-section be convex, prevents a tangent crossing the receiver boundary.

The concentrator collects radiation over an entrance aperture of width d_1 and an angular field of view of θ_{max} (half acceptance angle) in the plane transverse to the cylindrical concentrator. The rays incident at angles $\theta > \theta_{max}$ are reflected out of the concentrator. The maximum concentration that can be achieved by the modified Winston concentrator is [10]:

$$\frac{d_1}{S} = (1/\sin\theta_{max}) \qquad (2.6)$$

Where S is the circumference of the receiver pipe. This concentrator design could be successfully used where it is desired to illuminate or heat the entire outer surface of the receiver pipe. The height of the concentrator walls may be reduced substantially at one's convenience with very little loss of the entrance apecture and, hence, the

concentration ratio. The angular acceptance angle ($2\ \theta_{max}$), however, remains unchanged.

The Winston concentrator is of stationary type and requires only periodic adjustment depending upon the acceptance angle θ_{max}, concentration ratio (d_1/S), month of the year and the latitude of the place. The need to track the sun is, thus, eliminated.

In our case, the concentrator is required to collect the sun energy on its receiver surface at a temperature of 110°C at the rate of 543·75 kcal/hr in order to generate 1·78 kg/hr vapours of the methyl alcohol at 100°C (90°C saturation temperature + 10°C superheat) when the initial temperature of the saturated methyl alcohol is kept at 40°C. All the design values mentioned here, have been derived as per design analysis of the condenser discussed in section 2·2.

The design crieteria for the modified Winston's concentrator are:

1. The concentration factor must be such that the heat energy can be collected at the required moderate temperature i.e. about 110°C.
2. The acceptance time for solar radiation on any day of the year must be sufficient to perform the experiment with the system.
3. The height of the concentrator should not exceed the easily manageable proportions.

4. Length to height ratio of the concentrator must be large enough so that the end effects are minimized.

5. Available pipe of the standard size must be used as the receiver pipe.

With these criteria in mind, the following parameters were chosen for the concentrator design:

Half acceptance angle	=	5^o
Receiver pipe diameter (OD)	=	5·0 cm
Concentration ratio (CR)	=	5·0

From the Table 2·2 given by Winston [11] the acceptance time for the concentrator is $6\frac{1}{2}$ hrs/day.

As explained theoretically by Winston [10], the profile of the concentrator has been drawn geometrically using the following procedure:

Using the equation (2·6),

the entrance aperture $d_1 = \pi \times$ concentration ratio

$$= \pi \times 5 \times 5$$

$$= 78{\cdot}5 \text{ cm}$$

From figure 2·4, reproduced from Winston's work [10], the height/aperture ratio for $\theta_{max} = 5^o$ and CR = 5, comes out to be 1·085. Hence,

height h of the concentrator

(from the center of the receiver pipe to the tip of the concentrator wall)

$$= 1{\cdot}085 \times \text{aperture}$$

$$= 1{\cdot}085 \times 78{\cdot}5 \text{ cm}$$

$$= 96 \text{ cm.}$$

TABLE 2.2 Acceptance Time For Different Acceptance Angles

Half Acceptance θ_{max}	Max. Concentration factor	Acceptance Time in hour per day
7^o	8.2	7.4
6^o	9.6	7.0
5^o	11.4	6.5
4^o	14.3	6.0
3^o	19.1	5.3
2^o	28.6	4.4

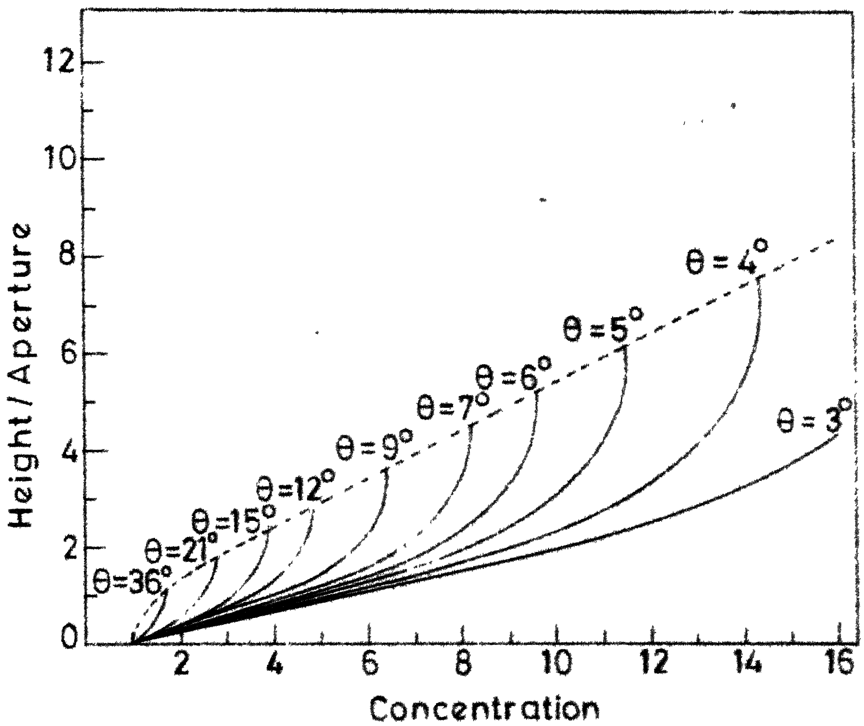


FIG. 2.4 HEIGHT/APERTURE RATIO VS. CONCENTRATION RATIO FOR DIFFERENT HALF ACCEPTANCE ANGLES (FOR FULL AND TRUNCATED COMPOUND PARABOLIC CONCENTRATOR)

Thus, the position of the point N_1, which is the tip of the concentrator wall, is fixed with respect to the pipe cross-section (figure 2·5). Let I_1N_1 be an incident radiant energy ray making an angle equal to the half acceptance angle θ_{max} (=5^o) with the vertical. Draw a tangent from the point N_1 to the outer diameter circle of the receiver pipe meeting at the point a. Thus, N_1a is a tangent on the pipe at a. L_1N_1 divides the angle I_1N_1a into two equal parts i.e.

$$I_1 N_1 L_1 = L_1 N_1 a$$

Now, the pipe diameter circle is divided into small sectors, each subtending an angle of 5^o at the centre and starting from ao (a small angle of 5^o is chosen to achieve higher accuracy). The corresponding points on the circle are b, c, d.....k, such that the last radius ko makes an angle of 5^o with the horizontal. A normal is then, drawn from N_1 to L_1N_1 which intersects the tangent on the point b of the circle at a point N_2 which is the second point of the concentrator profile. From the laws of reflection of light, it is known that any ray making an ~~an~~ angle $\theta < \theta_{max}$ with the vertical will reach the receiver pipe circle after being reflected by the surface N_1N_2. This ensures that all the radiant energy rays falling on the concentrator at an angle $\theta < \theta_{max}$ with the vertical will be reflected on to the receiver pipe surface. In order, now, to get the third point N_3 of the concentrator profile, a ray I_2N_2 is taken parallel to I_1N_1. Let L_2N_2 divide the angle I_2N_2b into two equal parts. A normal is drawn from N_2 to L_2 N_2 which intersects the tangent on the point c of the receiver pipe circle at

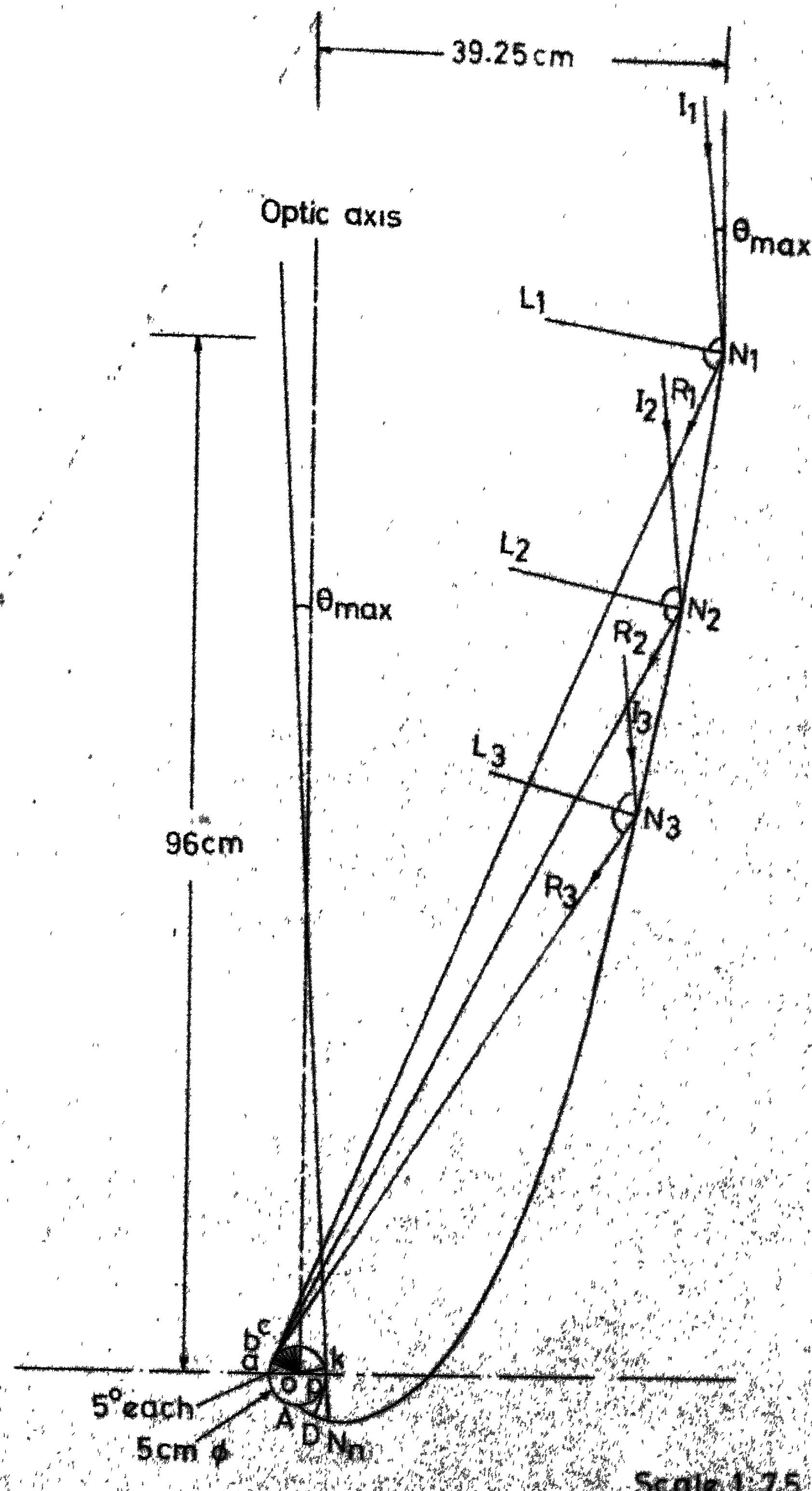


FIG. 2.5 PROFILE OF THE MODIFIED WINSTON CONCENTRATOR

a point N_3 which is the required third point of the profile. The process is continued till we reach the last point R on the circle and the corresponding point N_n on the profile.

The portion A N_n of the concentrator profile has to be the involute of a circle and is drawn by using the characteristics of the involute of a circle as follows:

Choose any point p on the arc of the circle between the points A and k and draw a tangent at the point p. Mark a point D on this tangent such that the length pD of the tangent = the arc length Ap. Thus, the point D lies on the involute of the circle. Similar points of the involute may be determined by choosing more points between A and k and they are all joined with N_n and A to complete the involute and, hence, the concentrator profile.

The actual concentrator profile, in our experiment, was drawn to the full scale and a plywood ''template'' was prepared. Mild steel flats were bent to the shape of the template and the buffed aluminium sheet was fixed by screws and nuts on these flats to acquire the shape of the modified Winston concentrator.

2·6 EVALUATION OF INPUT SOLAR RADIATION

If I_{GH} and I_{DH} are the instantaneous values of global and direct irradiation rates on a horizontal surface, we have

$$I_{DH} = 0.825\, I_{GH} \qquad (2.7)$$

The diffused radiation has been taken to be 17.5 per cent of the global solar radiation [12]. Theoritically, the diffused radiation is not concentrated on the receiver pipe by the concentrator. It is, therefore, neglected in computing the input solar radiation to the concentrator. The values of I_{GH} are taken from the Table 2.3, obtained from the Indian Meteorological Department [13], as the radiation measuring instrument was not available in the laboratory. The direct solar radiation at a tilted surface is given by

$$I_{DT} = I_{DH} \cdot \cos i / \sin \beta \qquad (2.8)$$

Where,

i = the angle of incidence that the direct solar radiation makes with the tilted surface,

β = altitude angle of the sun.

The angle of incidence is required to be calculated for the vertical side opening, facing either the East or the West and for the top opening of the concentrator which is tilted in the N - S direction.

For the vertical opening, facing either the East or the West,

$$\cos i = \cos \beta \cdot \cos (Z - n) \qquad (2.9)$$

TABLE - (2.5)*: Averages of Global Solar Radiation in cal/cm^2.

Station Kanpur 26.4° N latitude (Period of data Nov.'68 to Dec.'71)

Months/ hours.	06	07	08	09	10	11	12	13	14	15	16	17	18	19	Daily Total
JAN.		0.6	8.7	23.6	36.6	45.4	50.4	52.1	46.9	37.5	24.4	9.3	0.6		336
FEB.		2.0	14.8	31.6	44.9	56.1	62.2	61.5	55.1	44.9	30.9	14.0	1.6		420
MAR.		4.8	21.2	39.8	53.4	62.7	69.8	69.2	63.6	54.4	39.1	21.8	5.8	0.2	506
APR.	0.7	9.8	26.8	45.4	59.8	69.9	75.1	77.0	71.4	59.4	44.4	26.0	9.1	0.4	575
MAY	1.7	12.0	28.2	44.2	58.3	67.2	72.6	71.6	67.2	59.1	44.5	28.3	11.5	1.4	568
JUN.	1.9	9.8	22.9	36.5	44.7	55.1	65.8	63.1	56.7	47.7	36.6	24.1	10.9	1.3	477
JUL.	1.4	9.2	21.8	36.4	45.6	56.7	59.8	55.3	54.6	45.9	37.5	24.7	10.7	1.5	461
AUG.	0.6	8.0	19.8	30.9	39.4	47.2	47.8	47.5	47.0	44.0	32.8	21.9	7.6	0.5	395
SEP.	0.1	5.0	18.7	33.5	46.1	51.7	54.2	51.9	47.8	43.6	31.6	18.7	5.3	0.1	408
OCT.		2.9	16.8	34.1	49.2	59.4	63.5	62.0	59.0	47.5	33.0	17.1	3.4		448
NOV.		0.9	11.0	27.2	41.2	51.6	56.5	57.4	51.7	40.0	25.9	9.9	0.7		375
DEC.		0.4	8.2	23.1	37.1	47.4	53.7	53.6	48.1	37.8	23.0	7.9	0.3		341

* Table taken from India Meteorological Department, Meteorological Office, Poona-5.

where,

Z = azimuth angle,

n = wall solar azimuth.

For the top opening, incidence angle is given by

$$\cos i = \sin d . \sin (L-b) + \cos d . \cos (L-b) . \cos h \quad (2.10)$$

where,

d = declination of the sun,

L = latitude of the place,

h = hour angle,

b = angle of slope, that is the angle made by the sloping surface with the horizontal.

This is either positive or negative according as the tilt is towards the South or the North.

The solar energy enters the concentrator through its top and either of the end openings depending upon the time of the day. The rates of incident direct solar radiation (I_{DT}) are computed, knowing the angles of incidence for both the top and the end openings.

Hence, the total energy Q_{IT}, received by the concentrator is given by

$$Q_{IT} = I_{DTT} \times A_T + I_{DTE} \times A_E \quad (2.11)$$

Where,

A_T = top opening area,

A_E = end opening area

I_{DTT} = rate of direct radiation incident on the top opening and

I_{DTE} = rate of direct radiation on the end opening.

As mentioned above, the solar energy enters the concentrator through its top and either of the end openings depending upon the time of the day. However, the energy entering the concentrator through its end opening is small. Moreover, it is compensated by the loss of the energy through the other end. Hence, for the purpose of concentrator design, it is neglected and only the energy entering through the top opening is taken into account.

The averages of the global solar radiation on a horizontal surface, at Kanpur, are given in Table 2.3. The direct solar radiation, on a surface tilted by an angle of 41° 26' from the horizontal, calculated using equations 2.7-2.10 for the month of October 76 are given in Table 2.4.

Table 2.4 Monthly Average of the Total Solar Radiation on a Tilted Surface.

Time of the day / Month	Hourly Sum in cal/cm^2 for the hour ending at -					
	10.00 A.M.	11.00 A.M.	12.00 Noon	1.00 P.M.	2.00 P.M.	3.00 P.M.
October	52.5	61.8	65.5	64.5	63.0	52.6

The average of the direct solar radiation for the month of October '76 comes out to be equal to 60.0 cal/cm^2- hr on the tilted surface used in our experiment.

The heat required to generate the dry saturated vapours at 90°C at the rate of 1.78 kg/hr, when the initial temperature of the liquid saturated methyl alcohol is 40°C, is 512.55 k cal/hr (sec 2.2). Assuming, the efficiency of the concentrator to be 33 percent, the area of collection is given by

$$A_E = \frac{\text{Total heat required to be collected}}{\text{average direct solar intensity x collection efficiency.}}$$

$$= \frac{512.55}{60.0 \times 0.33} \ m^2$$

$$= 2.55 \ m^2$$

The aperture of the concentrator is 78.5 cm. Therefore the length of the concentrator, for the above area, is given by

$$l_i = \frac{2.55}{0.785} \ m$$

$$= 3.25 \ m$$

Heat required to superheat the saturated methyl alcohol vapours (initially at 90°C) by 10°C, is given by

$$Q_S = \text{Flow rate of methyl alcohol x average value of specific heat for vapour x temperature rise}$$

$$= 1.78 \times 0.80 \times 10 \text{ k cal/hr}$$

$$= 14.25 \text{ k cal/hr.}$$

Taking the collection efficiency 15 percent for this portion of the concentrator*, where superheating of the methyl alcohol vapours takes place, the area of collection required is given by

$$A_{su} = \frac{14.25}{60.0 \times 0.15} \ m^2$$

$$= 0.161 \ m^2$$

Hence, the length of the receiver pipe for this collection area is given by

$$l_2 = \frac{0.161}{0.785} \ m$$

$$= 20.5 \ cm$$

Thus, the total length of the concentrator comes out to be

$$L = l_1 + l_2$$
$$= (3.25 + 0.205) \ m$$
$$= 3.455 \ m$$

---

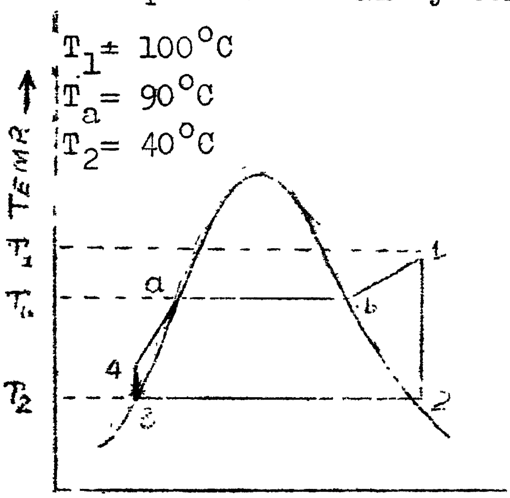
* The heat transfer coefficient between methyl alcohol vapours and the receiver is much less as compared to that between the liquid methyl alcohol and the receiver pipe. Also, there are more heat losses in the portion of the receiver pipe where superheating takes place because of higher surface temperature. Hence, the efficiency of this portion of the concentrator will be much less than the efficiency of the portion of the concentrator where boiling of methyl alcohol takes place.

The length of the concentrator, therefore, is taken 3.50 m.

The portion of the receiver pipe, where only vapours are present and not the liquid methyl alcohol, works as super-heater as well as accumulator.

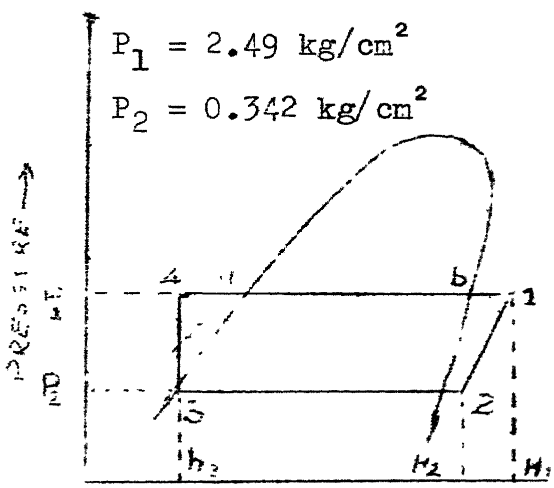
WORKING CYCLE

The working cycle of the present solar pump is the Rankine cycle, universally adopted for generation of power in a vapour or steam system.



Entropy →

Figure 2.6(a)



Enthalpy →

Figure 2.6(b)

The operating conditions and an ideal system T-Ø diagram is shown in figure 2.6(a). The process through the various components of the system are shown. 1-2 indicates the

expansion of vapour in the pump cylinder which utilizes some of the energy into pumping the water, 2-3 indicates the condensation of the vapour in the condenser, 3-4 is feeding of the methyl alcohol back to the evaporator, 4-1 represents the changes which occur inside the vapour generator (4-a is sensible heating of the liquid methyl alcohol, a-b is phase change from liquid to vapour, b-1 is superheating of the vapours) and, thus, the cycle is completed.

The same cycle represented on a P-H plane is shown in figure 2.5 (b)

CHAPTER III

MEASUREMENTS AND OPERATION OF THE SOLAR PUMPING SYSTEM

3·1 INSTRUMENTATION

(a) Temperature Measurement

All temperatures, except atmospheric temperature, have been measured by 24 gauge calibrated and insulated copper-constantan thermocouples. The receiver pipe, which also works as evaporator, has the thermocouples peened into its surface at 13 points along its length and spaced as shown in figure 3·1. Two thermocouples immersed in the fluid inside the evaporator pipe were used to measure the inlet and the outlet temperatures of the fluid flowing inside the evaporator pipe as shown in figure 3·2. Six thermocouples peened into the surface of the solar pump cylinder and the pipe connected to its lower end were used to get the temperature distribution of the solar pump cylinder and the pipe connected below it. The positions and spacing are shown in figure 3·3. Three more thermocouples, one at the outlet pipe for vapour of the solar pump cylinder and one each at the inlet and the outlet of the condenser copper coil were used to measure the temperatures at these points. The accuracy of all the thermocouples was checked and was found to be within $\pm 1 \cdot 0^{o}C$. An alcohol thermometer graduated from $0^{o}C$ to $110^{o}C$ with least count of $1^{o}C$, placed in the shade, was used to measure the ambient temperature. All the thermocouple temperatures were recorded directly on the graph paper by a 24 point Honey Well temperature recorder having the least count of $1^{o}C$.

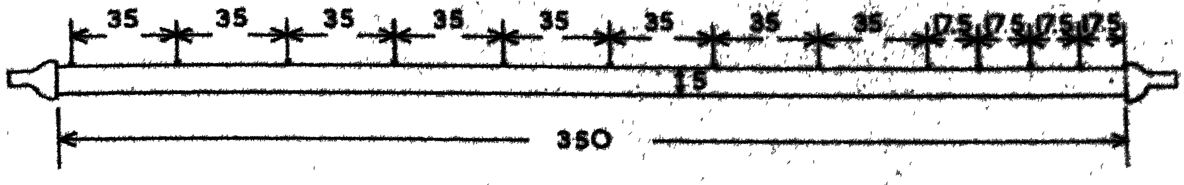


FIG. 3.1 LOCATIONS OF THE THERMOCOUPLES ON THE RECEIVER PIPE

All dimensions in cms.

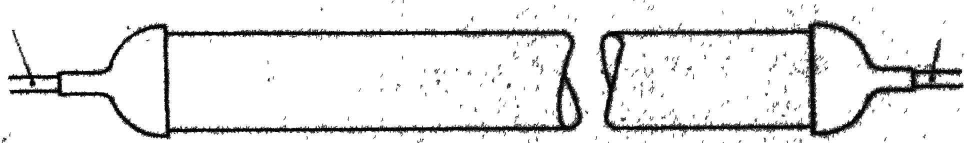


FIG. 3.2 LOCATIONS OF THE THERMOCOUPLES AT THE INLET & OUTLET OF THE RECEIVER PIPE

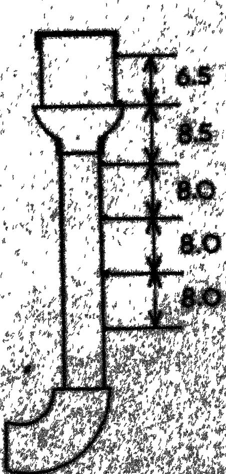


FIG. 3.3 LOCATIONS OF THE THERMOCOUPLES ON THE PUMP CYLINDER AND ITS LOWER PIPE

(b) Evaluation of Direct Solar Radiation.

The concentrators accept only negligible amount of diffused radiation. Hence, only the direct solar radiation was taken for the purpose of concentrator design. Due to non-availability of the solar radiation measuring instrument, the observed values of the global solar radiation for Kanpur published by the Indian Meteorological Department have been employed to compute the direct solar radiation. Table ?.3 gives the monthly averages for total irradiation on horizontal surfaces exposed to the sun on clear days, at Kanpur. The observations to assess these values were made between the years 1968 and 1973. The values check very well with corresponding measurements made by Chauhan [14] with an Eppley Pyranometer between March and May 1974.

The diffused radiation on a horizontal surface is estimated to be approximately 15-20 percent or an average of 17·5 percent of the total radiation received on clear days [12]. On the basis of this, the direct radiation on a horizontal surface has been taken to be 82·5 percent of the total radiation in calculating the collection area of the concentrator.

(c) Pressure Measurement

The pressure inside the evaporator pipe was measured with the help of a pressure gauge having calibrated scale both in kg/cm^2 and $lb/inch^2$. The range of the pressure gauge was (0·0 - 10·50) kg/cm^2 and although, the pressure gauge scale had divisions of 0·20 kg/cm^2, pressure upto 0·10 kg/cm^2 could be read fairly accurately.

The pressure gauge was mounted on the periphery of the evaporator pipe and at the end of the pipe at the higher elevation. This position of the pressure gauge had two advantages:

(i) The entry of any liquid methyl alcohol inside the pressure gauge was prevented. Only the vapours of the methyl alcohol come in contact with the pressure gauge.

(ii) The pressure gauge occupied a position just in front of the operator. This helped the operator to have a continuous watch over the pressure gauge reading and to maintain the optimum pressure inside the evaporator by varying the number of working cycles per unit time, while operating the inlet and outlet valves of the pump cylinder.

(d) Solar Pump Discharge Measurement.

A plastic bucket of 15 litres capacity, graduated in litres, was used to measure the discharge of the solar pump. Since the capacity of the bucket was small, a measured quantity of water was added into it periodically and was taken into account for the discharge readings.

3.2 OPERATION OF THE SOLAR PUMPING SYSTEM

Before starting the operation of the solar pumping system for testing, the following auxiliary operations were performed:

(a) Charging of the system with the working fluid.

(b) Priming of the solar pump.

(c) Removal of gases other than the methyl alcohol vapours present inside the system.

All these auxiliary operations are required only once with the system provided there is no leakage in the system.

(a) Charging of the system with the working fluid.

On the top of the storage tank, two ordinary pipe valves are provided to facilitate introducing the working fluid (methyl alcohol) inside the system. The other valves connected to the storage tank remain closed except these two valves. The methyl alcohol is poured through the one open valve and gases inside the storage tank escape out through the other open valve. Thus, the predetermined quantity of the methyl alcohol (about 5 litres) is introduced inside the storage tank and, then, these values are closed tightly.

(b) Priming of the solar pump:

A 1/2''diameter and 5 feet long polythene pipe was slipped over the suction pipe of the solar pump. Through this polythene pipe, the water was poured to go inside the solar pump cylinder, by keeping the other end of polythene pipe at a level higher than the solar pump cylinder, as the non-return valve in the suction pipe opens upward. The level of

the water in the pump cylinder was kept upto the lower seat of the float valve. Over the water level inside the pump cylinder, a fixed quantity (750 ml) of turpentine oil was poured through the valve provided at the top of the cover cap of the cylinder. This valve was, then, closed tightly. During the priming of the solar pump, the hand operated valves at the inlet and outlet of the pump cylinder for methyl alcohol vapours are kept closed.

(c) Removal of gases, other than the methyl alcohol vapours, present inside the system:

If the air or any other non-condensable gases are left inside the system, they get collected in the low pressure side i.e. in the condenser and the storage tank during the operation of the system. The presence of these gases in the condenser and the storage tank reduce the capacity of the condenser and decrease the vacuum inside the condenser and the storage tank. Hence, the removal of the air and other gases except the methyl alcohol vapours inside the system is necessary before starting the operation of the system. This is achieved by allowing the methyl alcohol vapours to pass through the condenser and the storage tank by opening both the hand operated valves. and to escape to the atmosphere through an open valve at the top surface of the storage tank. The air or any other gases in the evaporator, pump cylinder, condenser and the storage tank are, thus, driven out alongwith the methyl alcohol vapours. The following precautions are to be taken in doing so:

(i) there should be no cooling water flow in the condenser, otherwise, the methyl alcohol vapours will get condensed in it and will not be, therefore, able to drive out the air or gases inside the storage tank and the condenser.

(ii) The outlet valve of the pump cylinder for the methyl alcohol vapours and one valve on the top of the storage tank must be opened fully first and then the inlet valve to the pump cylinder must be opened gently and slowly, otherwise, the methyl alcohol vapours, which are at high pressure, will push the primed water in the solar pump out through the delivery pipe.

(iii) When the air or gases inside the system are removed, the inlet valve of the pump cylinder and the valve at the top of the storage tank must be closed simultaneously. The outlet valve of the pump cylinder is then closed. It should be checked that all the valves are closed air tight.

Once the above auxiliary operations are done, the system is ready to be operated provided the direct sun is available. It is desirable, however, that before starting the operation of the system, the following checks are made:

(i) The concentrator is given a required tilt (described in the appendix A) with the vertical so that it receives all the direct sun rays falling on it and concentrates on the receiver pipe for all working hours in a day.

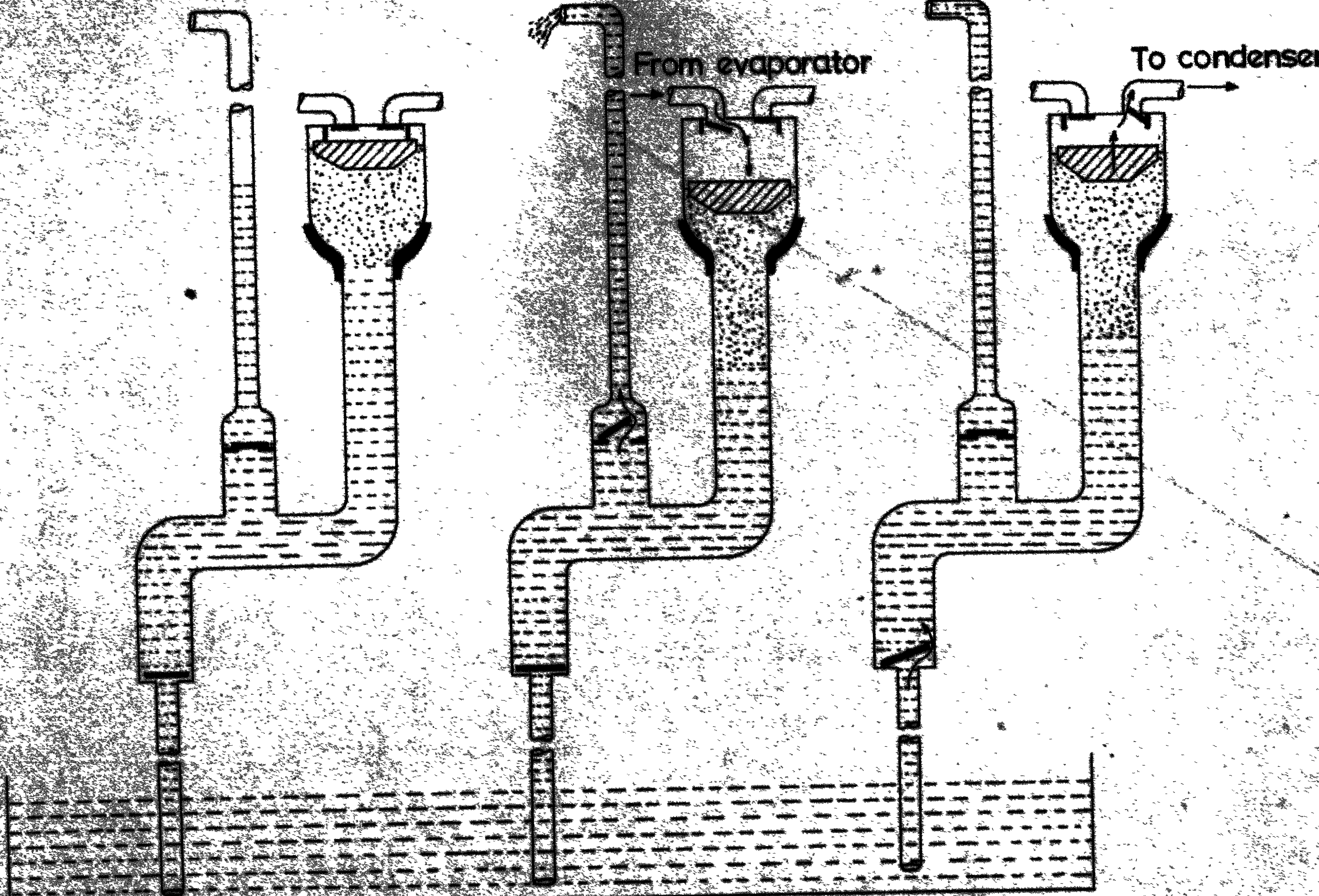
(ii) The dust collected on the glass cover and on the reflecting aluminium surface is removed with soft brush.

(iii) The suction pipe of the pump is dipped into the water to be pumped out.

Now, the operation of the system is easy. Initially, both the inlet and the outlet valves of the pump cylinder are closed. The inlet valve of the pump cylinder is first opened. The methyl alcohol vapours at a high pressure enter the pump cylinder and the pressure throughout the solar pump is increased. Due to this pressure, the non-return valve in the delivery pipe opens and the water goes out continuously through the delivery pipe, as more methyl alcohol vapours enter and expand, till the float valve rests on the seat to form a leak proof joint. The inlet valve is now closed and the outlet valve of the pump cylinder is opened. The vapours inside the pump cylinder go out to the condenser and get condensed. A vacuum is, thereby, created causing the non-return valve in the delivery pipe to close. Thus, the water in the delivery pipe would not be able to come back to the solar pump. The non-return valve in the suction pipe, simultaneously, opens and the water is sucked from the water tank into the solar pump. This continues till the upper surface of the float valve meets the ring welded in the cap-cover which serves like a leak proof joint. The outlet valve of the pump cylinder is then closed. The vacuum inside the solar pump is destroyed and the non-return valve in the suction pipe is closed due to its own weight so that the water from the solar pump can not go back to the water tank. The same cycle is repeated for continuous operation of the solar pump to lift the water. It is important that the pressure inside the evaporator is kept optimum* by adjusting the number of cycles per unit time.

The figures 3·4(a), 3·4(b) and 3·4(c) are self explanatory of the working of the solar pump.

·*· For a particular discharge head, we may have a range of the pressure inside the evaporator pipe. The pressure at which the pump works best is given the name ''optimum pressure'' and is decided by operating the solar pump at different evaporator pressures. For a discharge head of 5·6 meters, the optimum pressure inside the evaporator was found to be about 2·0 kg/cm^2 (gauge).



Primed pump (a) | Delivery stroke (b) | Suction stroke (c)

CHAPTER - IV

4.I RESULTS AND DISCUSSION

The testing of the experimental model was carried out in two phases:

In the first phase, the experiments were conducted only with the concentrator used in the system to study its performance. Although, tests were carried out for a week for different flow rates of water and inlet and outlet temperatures, only the results obtained on June 20 and June 23, 1976 are reported and plotted in figures 4.1 and 4.2 as representative results.

The curves shown in these figures indicate the variation during the test hours in a day in the inlet and the outlet water temperatures, the temperature rise of the water flowing in the receiver pipe and variations in the efficiency of collection of the concentrator. It should be noted that in figure 4.1 (June 20, 1976), the temperature of the inlet water indicates the temperature of water which is recirculated and, hence, the inlet temperature values are so high. In figure 4.2, however, fresh inlet water temperature has been plotted.

The outlet water temperature and the temperature rise curves in both the figures are seen to reach peak values between 12.00 A.M. - 2.00 P.M. It is because, during this time the sun's rays are in the East-West plane containing the optic axis of the concentrator, so that the rays after being reflected from the reflecting surface, fall on the

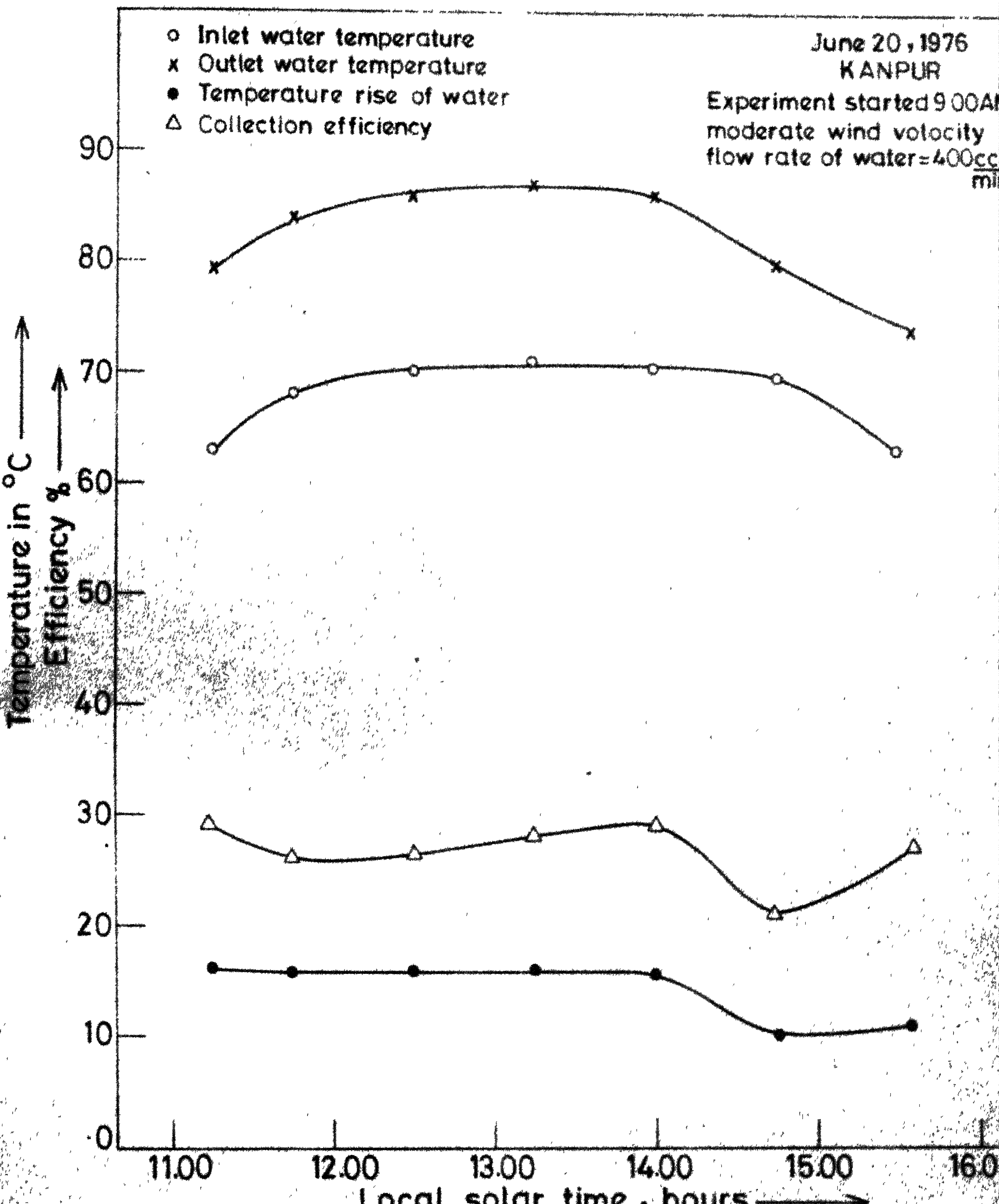


FIG.4.1 EXPERIMENTAL TEST RESULTS FOR CONCENTRATO

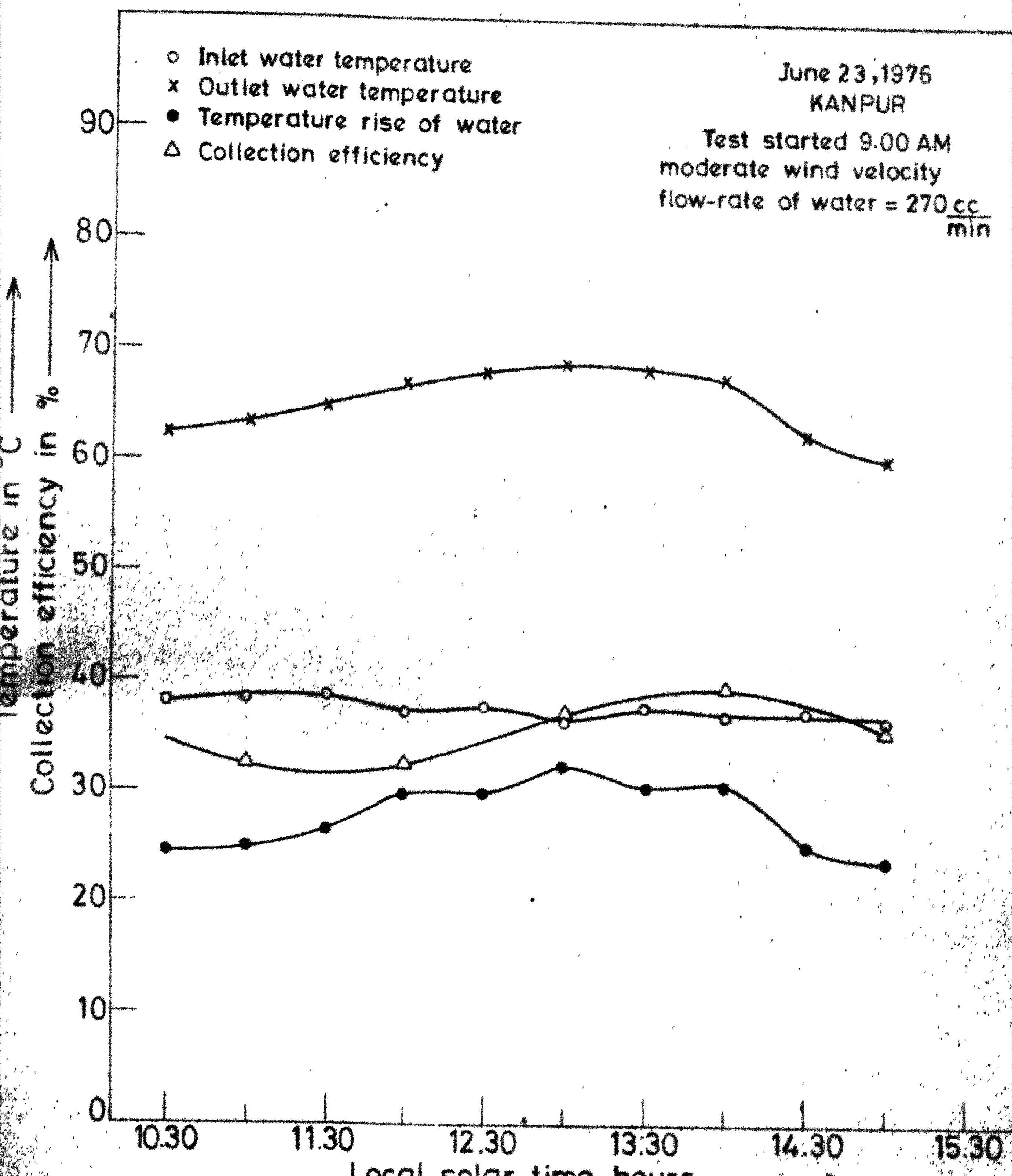


FIG.4.2 EXPERIMENTAL TEST RESULTS FOR CONCENTRATOR

receiver pipe making maximum angle with the tangent at that point and, hence, the absorptivity is high. Moreover, the end effects during this period are minimum and the solar radiation falling during this period is also maximum. Thus, in-spite of higher heat losses (due to high absorber temperature) the temperature rise and, hence, the outlet temperatures of water are high during this time.

The collection efficiency of the concentrator is seen to be maximum at about 2.00 P.M. in both the figures. One of the reasons for the same is that the concentrator axis is tilted in the East-West direction at a small angle with the horizontal. Hence, the sun rays are normal to the concentrator not at 12 noon but some time later in the afternoon.

The efficiency curves in the two figures show a sudden fall and rise in their behaviour at certain times of the day. This is because of the sudden clouds appearing at these times obstructing the direct sun radiation.

After performing the above tests, further tests were taken up when the whole pumping system was ready in October 1976.

The figure 4.3 shows the temperature distribution of the receiver pipe surface at various hours of the day when no working fluid is flowing through it, on October 31, 1976. In the early hours of the day (about 9.00 A.M.), the temperature reaches the peak values at short distance from the western end of pipe, whereas, in the afternoon, the pipe temperature is seen to be maximum near the eastern end of the receiver pipe. Since, the water flows through the

concentrator from the west to the east direction, the heat transfer between the receiver pipe and the water flowing through it could be compared with the parallel and counter flow type of heat exchange in the morning and afternoon hours, respectively. This, therefore, helps to explain the higher collection efficiencies obtained during the afternoon hours, around 2.00 P.M., compared to the values at the solarnoon.

In the second phase, the tests were conducted with the complete pumping system. The basic idea of such tests was to check the feasibility of the working of the theoratical model of the pump under study. The system was run for several days and the pumping of the water was possible from about 9.45 A.M. to 3.15 P.M. The discharge rate of water varied with the time of the day depending upon the output[*] of the concentrator and, hence, the solar radiation intensity. The discharge of the pump was not continuous as the pump is single acting reciprocating type. The representative results of these tests have been presented in figure 4.4 - 4.7.

Figure 4.4 shows the pressure - temperature diagram for the commercial methanol as compared with the pressure - temperature diagram for pure methyl alcohol, as taken from the Handbook [8]. The agreement between the two diagrams is quite satisfactory. The small deviations in the two curves are due to the following reasons:

(1) The experiment has been performed for the commercial

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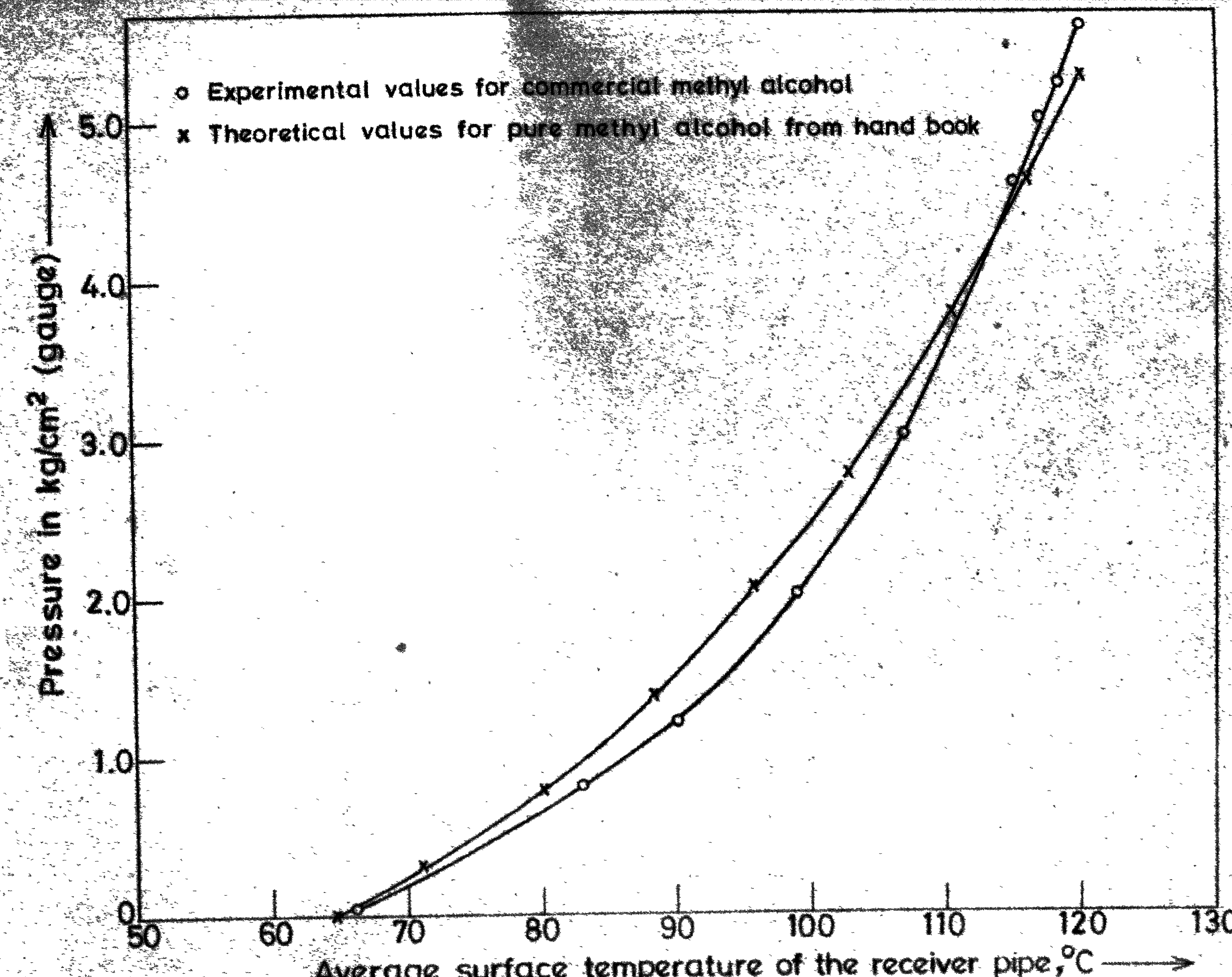
*. Output of the concentrator is defined as its capacity of generating methyl alcohol vapours per unit time.


o Experimental values for commercial methyl alcohol
x Theoretical values for pure methyl alcohol from hand book
Pressure in kg/cm² (gauge)
5.0
4.0
3.0
2.0
1.0
0
50
60
70
80
90
100
110
120
130
Average surface temperature of the receiver pipe, °C

methanol having impurities.

(2) The temperature readings have been taken on the outer surface of the pipe and not of the fluid inside the receiver pipe.

It is expected that the fluid temperature inside the pipe will be about 3-5^{o}C less than the surface temperature.

In figure 4.5, the pressure generated outside the receiver pipe by methanol vapour is plotted against the time of the day for November 1, 1976. It is seen that the pressure increases as the time increases. This information is helpful to guess the time when sufficient pressure is built up to start the operation of the pump.

The discharge of the pump versus time of the day is plotted in figure 4.6 for the experiment performed on November 9, 1976. Also, the estimated values of the discharge rate (swept volume x No. of cycles per unit time) vs the time of the day is plotted in the same figure. The discharge of the pump is increasing in the morning hours and reaches the maximum value between 12.00 noon - 12.30 P.M. It is because the solar radiation intensity is increasing in the morning hours and starts decreasing in the afternoon. Therefore, the methanol vapours available for operating the solar pump are maximum in quantity just in the afternoon, and hence, more number of cycles per unit time can be completed at this time as compared to those in the morning hours or in the late afternoon hours. The discharge rate values estimated theoretically (swept volume x number of cycles per unit time) do not agree with the experimental

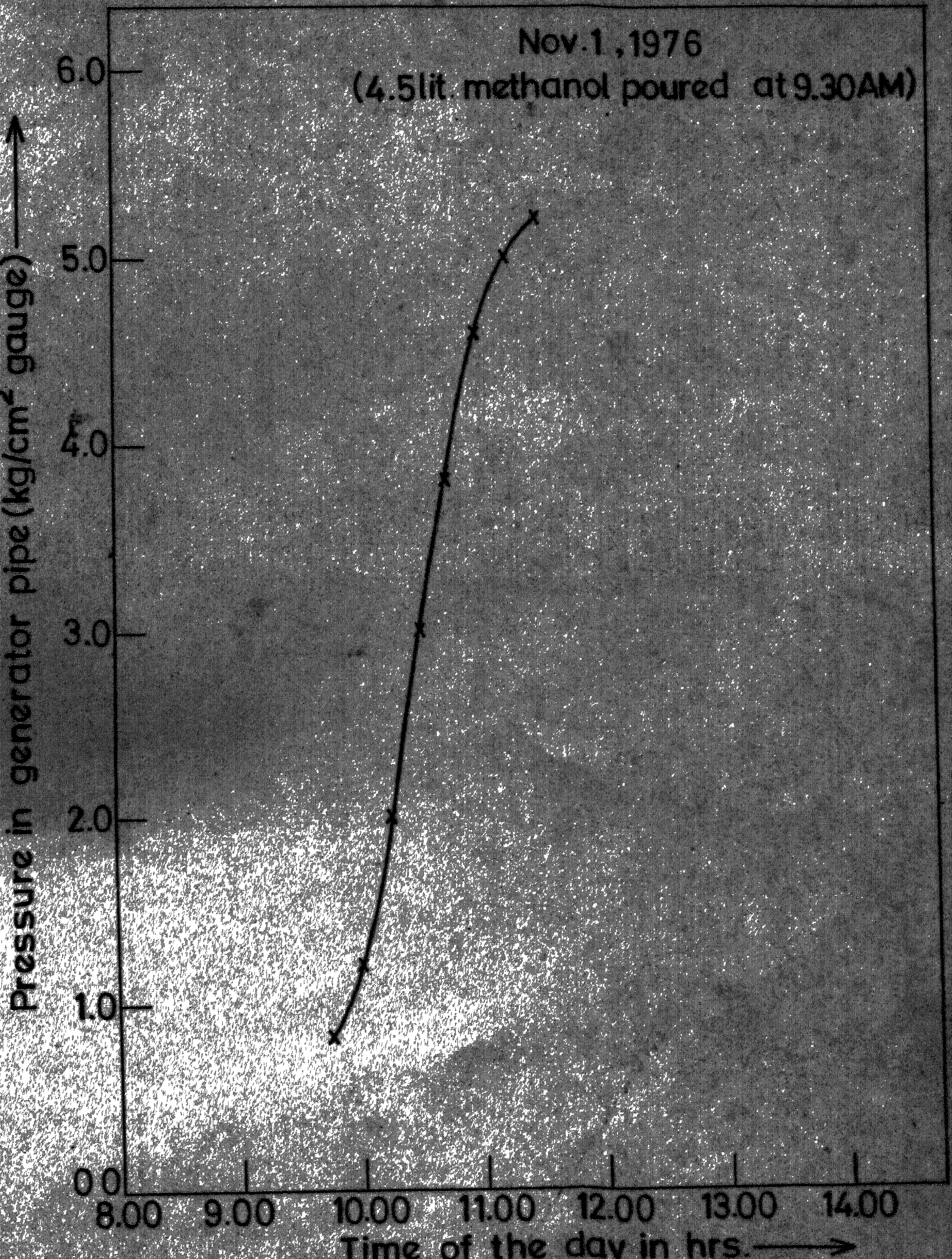


FIG.4.5 PRESSURE BUILT-UP IN SIDE RECEIVER PIPE VS. TIME OF DAY PLOT

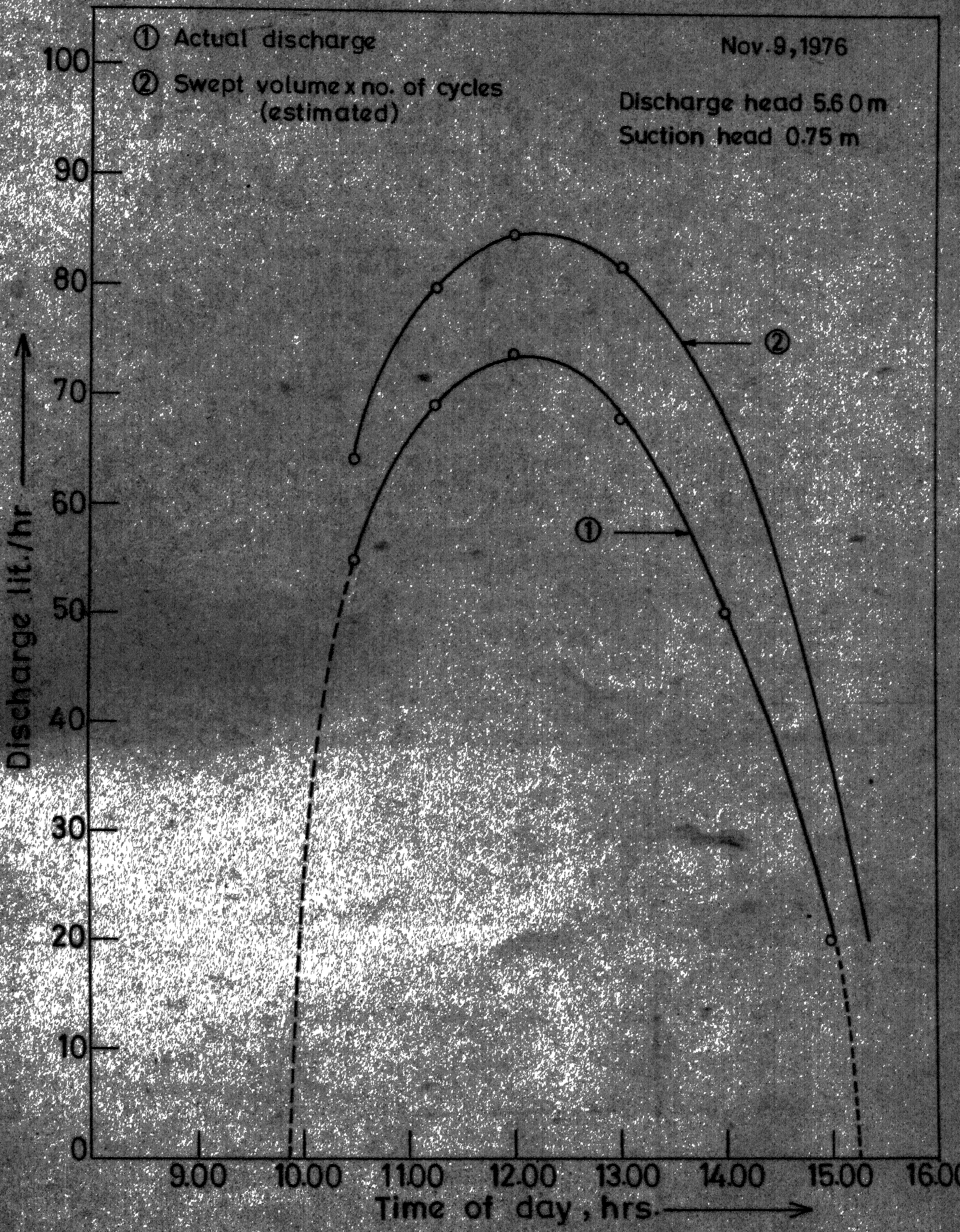


FIG. 4.6 DISCHARGE OF PUMP VS. TIME OF THE DAY PLOT

discharge rate values due to the fact that one-way valves used for water inlet and outlet purposes are not cent per-cent efficient.

The representative curve of the temperature distribution on the surface of the pump cylinder and its lower pipe is shown in figure 4.7. The readings were taken on November 9, 1976 at 11.15 A.M. At the upper end of the pump cylinder, the temperature is about 60°C because the hot vapours are first contacting this part of the cylinder and transfer most of their heat to this portion. At the end of the suction stroke, the turpentine oil is not reaching the upper-most portion of the pump cylinder due to the presence of the float valve. In the portion of the lower pipe attached to the pump cylinder where methanol vapours and hot turpentine oil are coming into contact alternatively, the temperature is about 52°C. In the lower pipe in which the hot turpentine oil and cold water are coming into contact, the temperature of the surface is slightly higher than the temperature of the cold water.

After the experiment was completed, the fluids charged inside the pump cylinder were taken out and it was found that there was no trace of methyl alcohol inside the pump cylinder. This ascertains that the temperature gradient inside the pump cylinder was small and no condensation of methanol vapours could take place there.

4.2 CONCLUSION AND SUGGESTIONS FOR FURTHER WORK

A new version of free, liquid-piston solar pump has been

developed which works on the Rankine cycle. It can lift water from deep levels, comparable to any theoretical estimates, and is capable of pumping it to high heads depending upon the design parameters of the solar collector and the properties of the working fluid used. The idea that has been tried is simple and works quite successfully compared to any other solar pump, reported so far. The entire cost of the development of the laboratory model of this pump is about Rs. 3805/- as detailed in Table 4.1.

TABLE 4.1 Cost Estimation Of The Laboratory Model System Developed.

S.No.	Item	Size	No./Qty.	Cost in Rupees
1.	Solar Pump		1	650
2.	Condenser		1	200
3.	Storage tank	10.8 lit	1	100
4.	Feed Pump		1	425
5.	Concentrator	$2.75 m^2$ Area	1	1000
6.	Water tank		1	30
7.	Methanol		10 lit	200
8.	Piping and Accessaries			300
9.	Fabrication and Labour			900

Total = 3805/-

The simple features of the design provide scope for developing a prototype of a working solar pump of the field-size at a reasonable cost.

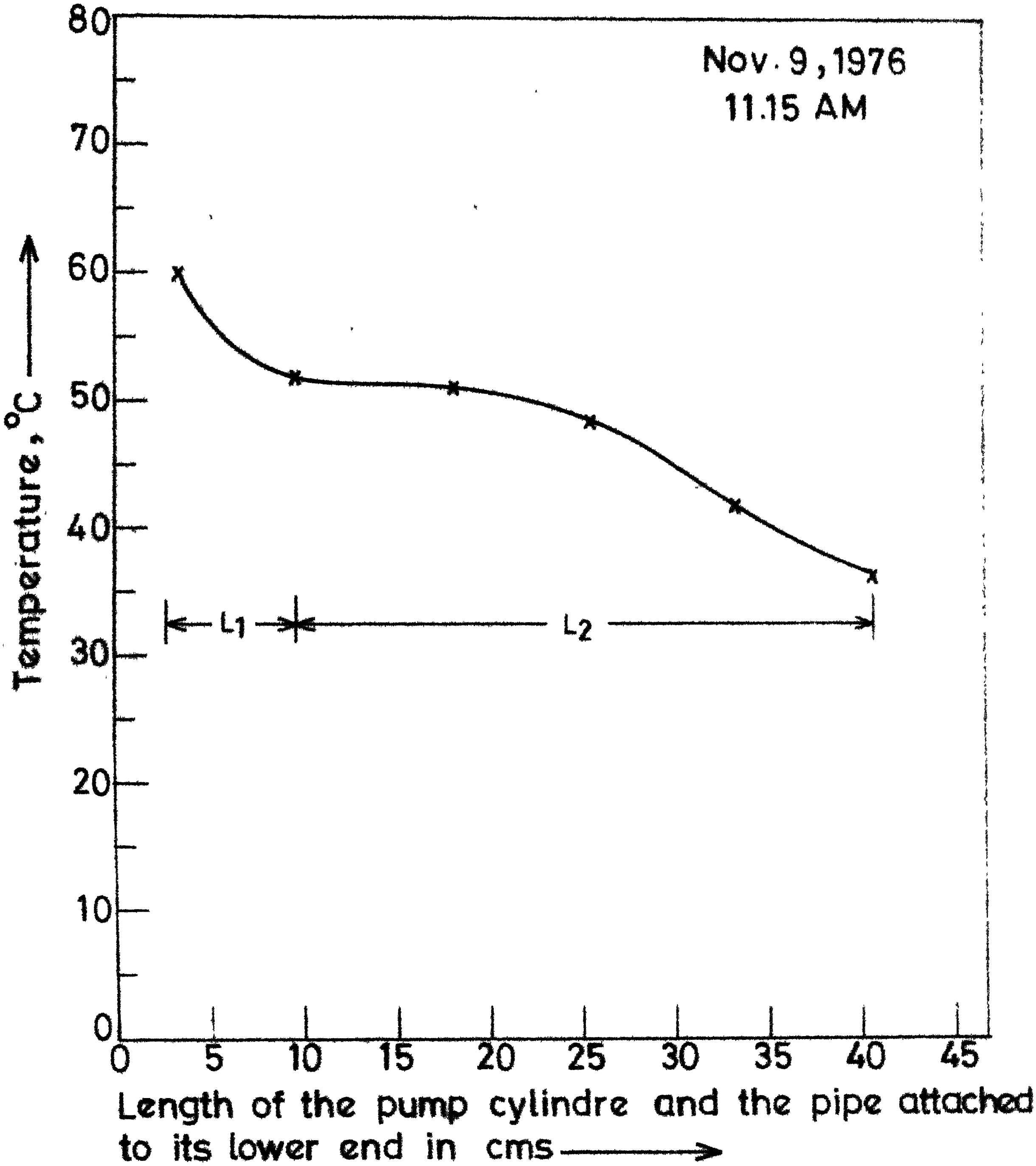


FIG. 4.7 TEMPERATURE DISTRIBUTION ON THE PUMP CYLINDER AND ITS LOWER PIPE

Based upon the experience gained in developing and testing the model, there seem to be many shortcomings which need improvement, modifications and alterations in the following directions:

1. The centrifugal feed pump used in the present work may be replaced by a small positive displacement pump such as gear pump and vane pump.

2. An accumulator must be used in between the solar pump and the concentrator receiver pipe to store the vapours. This will reduce the pressure fluctuations inside the receiver pipe when the solar pump is in operation.

3. The water lifted by the solar pump must be passed through the condenser to work as cooling water for the condenser. This will eliminate the need of cooling water circulation pump.

4. The suitable working fluids for different discharge and suction heads may be selected to give the best performance for a particular discharge or suction head.

5. The liquid piston may be replaced by a solid piston. This will eliminate the possibility of mixing of liquid piston in the fluid being lifted and thus getting discharged through the delivery pipe.

6. The inlet and outlet valves for working fluid vapours to the solar pump cylinder may be replaced by the automatic valves which could be operated by pressure difference available between the evaporator and the condenser.

REFERENCES

1. R.C. Jordan and W.C. Ibele, ''Mechanical Energy from Solar Energy'', Proceedings of the World Symposium on Applied Solar Energy, November 1-5, 1955, pp 81-101.

2. Holden, F. Charles, ''Solar Motors'', Scientific American, March 1901, p-169.

3. M.L. Khanna, ''Studies on Hot-Air Engine Run With Solar Energy Conducted At The National Physical Laboratory, New Delhi And a Critical Review Of Its Present Status'', Paper No. 7/14, International Solar Energy Society Conference, 1970.

4. J.P. Girardier and H. Masson, ''Utilization of Solar Energy for Lifting Water'', Paper No. 6/116, International Solar Energy Society Conference, 2nd-6th March 1970.

5. F.A. Bonaventura and V.E. Plymton, ''Solar Energy Pump'', U.S. Patent 2688922, 1954.

6. D.P. Rao and K.S. Rao, ''Solar Water Pump For Lift Irrigation'', Birla Institute of Technology and Science, Pilani, Rajasthan, India.

7. C. West, ''The Fluidyne Heat Engine'', Paper No.5, International Solar Energy Society, July 1974.

8. J. Timmermans, ''Physico-Chemical Constants of Pure Organic Compounds'' Elsevier Publishing Company, Inc. New York Amsterdam London Brussels, 1950.

9. D.Q. Kern, ''Process Heat Transfer'', Ist Edition, N.Y., Mc Graw-Hill, 1950.

10. R. Winston and H. Hinterberger, ''Principles Of Cylindrical Concentrators For Solar Energy'', Solar Energy, Vol.17, No.4, September 1975, pp. 255-258.

11. R. Winston, ''Principles Of Solar Concentrators Of a Novel Design'', Solar Energy, Vol.16, No.2 October 1974, pp. 89-95.

12. Central Building Research Institute, Roorkee, ''Climatological and Solar Data for India''. Second Edition, Sarita Prakashan, Meerut, 1969.

13. India Meteorological Department, ''Normals of Hourly And Daily Global Solar And Diffuse Solar (Sky) Radiation in cal/cm^2 Based on Data upto 1968''.

14. R.S. Chauhan,''Fabrication and Testing of a Collector-cum-storage Type of Solar Water Heater'' a M.Tech. Thesis, I.I.T. Kanpur, India, July 1974.

15. L.S. Cheema and K.D. Mannan, ''Analysis of Stationary Concentrators'', paper presented at Seventh Annual Conference of All India Solar Energy Working Group, Punjab Agriculture University, Ludhiana, India, November 13, 14, 1975.

APPENDIX - A

ANGLE OF TILT OF STATIONARY CONCENTRATOR

The tilt angle of the stationary concentrator is to be periodically adjusted according to the variation of the apparent position of the ecliptic plane of the sun during the year which causes a variation in the East-West vertical angle. The East-West vertical angle, EWV, defined below, is the primary factor in designing stationary concentrator and for computing the periodicity of its adjustment.

The East-West vertical angle, EWV, is defined as the angle of the sun's rays projected on a vertical N-S plane and is given by the relation:

$$\tan EWV = \tan \beta \tan Z \qquad (A\text{-}1)$$

Where,

β = altitude angle

Z = azimuth angle

The East-West vertical shift angle, V, is the shift in the EWV angle from the equinox position and can be computed from the following equation [15]:

$$\tan V = \tan d \tan h \qquad (A\text{-}2)$$

Where,

d = declination of the sun

h = hour angle.

If the value V, for $\pm$ h hours, calculated from this relation exceeds the total acceptance angle of the concentrator (10^{o} in the present design), then the tilt of the concentrator is changed.

THE ANGLE OF TILT (α)

Angle that the optic axis of the concentrator makes with the vertical, called the angle of tilt of the concentrator, can be computed by the relation:

Tilt angle = declination of the sun + latitude of the place + half acceptance angle.

i.e.

$$\alpha = d + L + \theta_{max} \tag{A-3}$$

However, near the equinoxes, the tilt of the concentrator may be kept equal to the latitude for symetric concentration. This can be visualized because the East-West vertical shift tends to zero during this period.

The adjustment of the tilt angle is required when the value of V exceeds the total acceptance angle ($2\theta_{max}$) for the specified $\pm$h hour of ddsired collection period. It is done by making use of equation A-3.